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Harry L. Horning

HARRY L. HORNING died January 4, at the Battle Creek Sanatorium.

The profession of engineering throughout the world, the automotive industry, the Society and his multitude of friends

suffer in his passing a loss whose extent and intensity cannot this early be clearly envisioned. His sphere of influence, growing out of a combined vigor and sensitiveness of spirit, was at once broad and intimate.

An enthusiast in his profession, Mr. Horning took an artist's delight in perfection of design, and contributed vastly to the development of the internal combustion engine.

Industry frequently called on Mr. Horning to represent it in Government activities. During the war he was chairman of the Automotive Products Section of the War Industries Board, Council of National Defense. As president of the Internal Combustion Institute he was active in the N.R.A. code authority representing manufacturers of the automotive type of Diesel engines; and was a member of the National Conference on Street and Highway Safety and of the Society's Military Motor Transport Advisory Committee (1931-1934).

S.A.E. President in 1925, and member of many committees, past and present, Mr. Horning served the Society since 1910. Especially valuable have been his contributions to its research activities, notably in the field of fuels. He was a major force in the movement to bring the automotive and petroleum industries together to deal with the fuel problem and in 1920 was the chairman of the Committee on Utilization of Present Fuels in Present Engines. He was an active member of the Research Committee, the Fuels Research Subcommittee and the Cooperative Fuels Research Committee since 1926. The present C.F.R. Engine was developed and is being built at the plant of the Waukesha Motor Co., and its recognition abroad was actively furthered by Mr. Horning's many foreign contacts.

As president and general manager of the Waukesha Motor Co., he was a capable executive and head of a family of 1300 plant employees. His humor and humanity, eager interest, clear sincerity, great courage, vitality and profundity

of spirit, and his sensitive appreciation of beauty won him the love and respect of his intimates.

He is survived by his wife and a sister.

Mr. Horning was 55 years old. He secured his early education at Carroll College Academy and Carroll College, both at Waukesha, Wis. He was associated for brief periods with the Milwaukee Gas Light Co., the Crane Co. and the Modern Steel Structural Co. In 1905 he established the Waukesha Motor Co., of which he was president and general manager.

Mr. Horning joined the Society in 1910. Besides the offices already mentioned, he was first vice-president during 1921 and chairman of the Research Committee in 1924 and 1929. He was the chairman of the Tractor Division of the Standards Committee and chairman of the Membership Committee in 1926; a member of the Finance Committee in 1926 and 1928; of the Motor-Vehicle Activity Committee in 1929; of the Diesel-Engine Activity Committee in 1931; of the Coker F. Clarkson Memorial Committee in 1931; and of the Joint SAE-ASME Diesel Fuel Research Committee in 1932 and 1933. He was S.A.E. representative on the National Screw Thread Commission foreign trip in 1919, and on the National Research Council, Engineering Division in 1928. He presented numerous papers at Society meetings, principally on tractor engines, fuels and research problems.

Mr. Horning was president of the Motor and Accessory Manufacturers Association in 1927. He was a member of the Institution of Automobile Engineers, of the American Society of Mechanical Engineers, of the Association for the Advancement of Science, and a Fellow of the Royal Society of London.

Engine Roughness— Its Cause and Cure

By P. M. Heldt

Engineering Editor, *Automotive Industries*

ROUGHNESS in the operation of engines has increased in seriousness with increase in the compression ratio and in the provisions for inducing turbulence in the combustion chamber, both of which factors tend to increase the rate of pressure rise in the engine. In this paper the thesis is maintained that this roughness consists of synchronous transverse vibration of the crankcase, due to variations in gas pressure and inertia forces. By synchronous vibration is meant a vibration which passes through a cycle in exactly the same time as the periodic force which produces it, so that the amplitude of the vibration builds up from cycle to cycle until the damping forces become equal to the exciting force.

Owing to the angularity of the connecting rod in all except the dead-center positions, the gas pressure produces an alternating horizontal force on the crankcase at the main bearings. This alternating horizontal force may be resolved into an endless series of harmonics of successively increasing frequencies. The actual frequencies naturally rise and fall with the speed of the engine, and when one of the lower harmonics corresponds in frequency to the natural frequency of the crankcase as an elastic vibrating body, serious lateral vibration is set up, which we know as engine roughness.

Gas pressures tend to cause lateral vibration in a horizontal plane only (in a vertical engine), but inertia forces tend to produce both horizontal and vertical vibrations. For each harmonic of the horizontal vibrating force except the first there is an harmonic of the vertical vibrating force, and if there happens to be approximate phase equality between the horizontal and vertical harmonics they will combine to produce vibration in an inclined plane. As the plane of vibration shifts from the horizontal, the natural frequency of vibration of the crankcase is likely to increase gradually, which is probably one reason why roughness generally is not confined to a narrow speed range but covers the whole range above a certain minimum critical speed.

The best safeguard against trouble from rough operation is to so design the crankcase that it is quite stiff in both the horizontal and vertical planes.

As the stiffness increases, so does the natural frequency of vibration of the crankcase, and if the latter is sufficiently stiff, only the higher harmonics of the transverse forces can come into synchronism with it. These higher harmonics are of such small magnitude that they cannot do much harm.

VIBRATION has been a problem of engine designers from the very beginning, but the type of vibration that caused the greatest trouble has changed from time to time. In the earliest engines, with a small number of cylinders, the chief cause was lack of balance of reciprocating parts. The engines at that time were rigidly mounted on the car frames; at certain speeds the impulses due to inertia of

reciprocating parts came into resonance with the frame, and a sort of frame quiver resulted which was quite unpleasant. Later, trouble began to be experienced from torsional vibration of crankshafts. This naturally occurred most frequently with long shafts, because the torsional flexibility of the shaft increases with its length. As a matter of fact, this form of vibration became a cause for worry with the advent of the six-cylinder engine. This type was exploited first by Napier in England, which may explain why the first torsional vibra-

[This paper was presented at the Annual Meeting of the Society, Detroit, Jan. 16, 1936.]

tion damper for automobile engines was developed in England—by Lanchester. That was about 1914.

A decade later we became conscious of vibrations due to torque reaction. That cyclic torque variations caused the engine to vibrate had been known for a long time, but up to the later twenties this had been taken for granted. There can be no torque without an equivalent reaction, and since the torque itself is the result of a quick succession of impulses, the reaction also must be intermittent. With modern multi-cylinder engines torque-reaction vibration is perceptible only at very low speeds, particularly when the engine is cold, and flexible engine mountings have practically eliminated this form of vibration from the frame and body.

In recent years we have heard a great deal about engine roughness. So far as I am aware, the use of this term in the literature of the internal combustion engine is not more than 10 years old, though it may have been used earlier in testing plants. "Roughness" evidently was a different type of vibration from the original type due to unbalanced reciprocating parts, from torsional vibration, and from the form of

high-frequency vibration known as gas-knock or detonation, which also had been known since the earliest days of the internal combustion engine. Roughness became a serious factor about the time the high-turbulence combustion chamber was introduced and the simultaneous shift to six- and eight-cylinder in-line engines, both of which are relatively long. At first the nature of this particular kind of vibration was quite obscure, and I remember asking the author of a paper on combustion phenomena, who repeatedly used the term, a number of questions about it. I asked him how he knew that the trouble was not due to torsional vibration, and his reply was that, unlike torsional vibration, it was not confined to narrow speed ranges but was present throughout a wide speed range.

It did not take long, however, to discover that the phenomenon known as roughness had something to do with the rate of combustion and, therefore, with the rate of pressure rise in the cylinder and also with the flexibility of the crankcase. I believe it was Ricardo who first pointed out that if the rate of pressure rise in the combustion chamber was not more than 30 lb. per sq. in. per degree of crank travel, the engine was likely to be smooth, whereas with considerably higher rates of pressure rise it would be rough. That phase of the problem has been attacked mainly from the chemical angle. Fuels have been developed which under given conditions burn less rapidly and therefore do not give rise to abnormal rates of pressure rise. The pressure rise, of course, is the thing that excites the vibrations. There is another, contributory factor, however, and that is the flexibility of the body which responds to the exciting cause—in this case the engine block or the crankcase. If the crankcase is lightly built and especially if it is not very rigid in the directions in which the exciting forces tend to deflect it, then it will respond readily to these exciting forces. What is particularly important is that if the crankcase has little rigidity, it can vibrate at a low frequency and therefore respond to impulses of

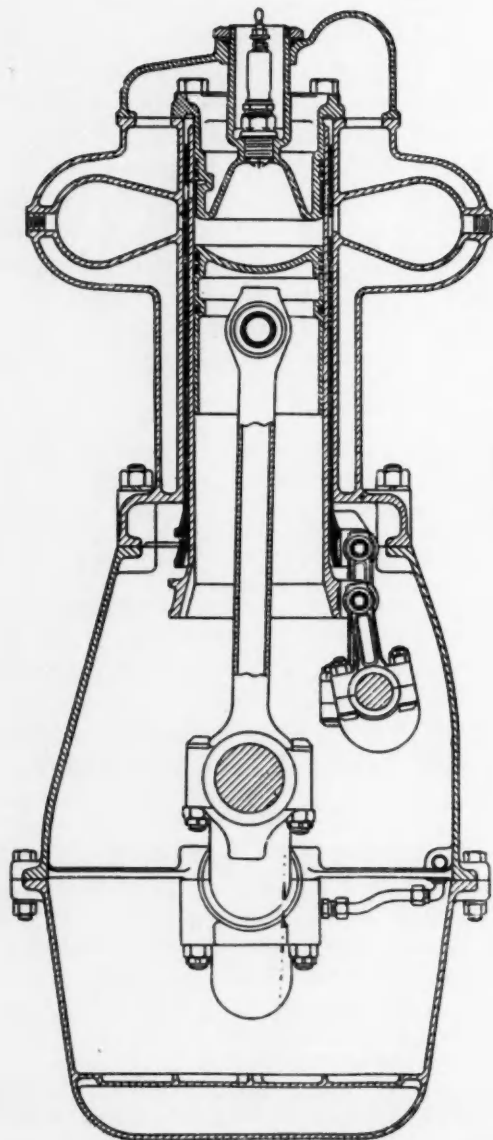


Fig. 1—(left)
Knight Engine
with Crankcase
of Little Horizontal
Rigidity

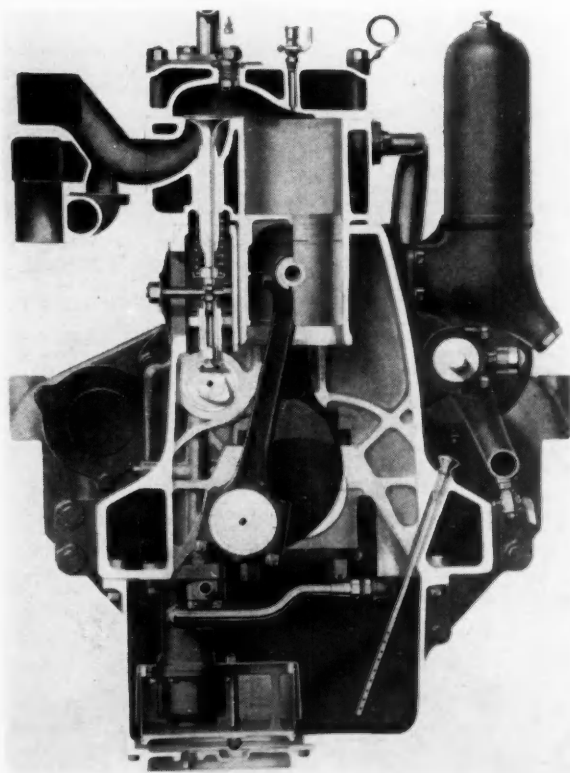


Fig. 1—(right)
Waukesha Bus
Engine with
Crankcase of
Great Vertical
and Horizontal
Rigidity

lower frequency, which impulses, as will be shown later, are always of the greatest magnitude.

Engines are generally supported at the ends, and any impulse received by the crankcase anywhere between supports in a direction transverse to the crankshaft axis naturally tends to deflect the crankcase transversely. It is somewhat questionable, however, whether the exact location of the supports has much to do with the tendency to roughness, because in modern engines, in order to prevent imparting vibrations to the car frame and body, the supports are usually of the flexible type. The chief resistance to the impulses undoubtedly comes from the inertia of the masses at the ends of the crankcase. At the rear end there are the flywheel, clutch, and clutch housing, which together make a very respectable mass even in a multi-cylinder engine; and at the forward end there are the timing gears and their housing, and possibly other parts rigidly mounted on the crankcase, which also constitute a considerable mass.

That smooth operation, particularly of high-speed engines, requires rigidity in the crankcase has been known for a considerable time now, but it was not always realized, and it is interesting to compare some of the earlier designs with more modern ones from this point of view. In Fig. 1, the view on the left shows the crankcase design of a Knight sleeve-valve engine built in considerable numbers about 1914. It will be seen that the crankcase is a mere shell, showing practically no evidence of any attempts to stiffen it against deflection by the forces due to gas pressure and the inertia of the moving parts. It is true that the upper section of the crankcase is comparatively deep, but the reason for making it so undoubtedly was that room had to be provided within it for the mechanism for reciprocating the sleeves and not with any idea of increasing the vertical stiffness of the case. These engines had only four cylinders, hence the crankcases were short, and as the speed was limited to about 1800 r.p.m., they performed satisfactorily, considering the requirements of the times.

On the right is shown the crankcase of a Waukesha bus engine of recent design, which appears to be an extreme in respect to provisions made to insure rigidity. It is true that this crankcase is of aluminum, whereas that of the Knight engine was of cast iron; aluminum has a somewhat smaller modulus of elasticity than cast iron and this must be compensated for by more massive design. The great depth of the crankcase will be noted, giving great stiffness in the vertical direction, as will also the heavy ribbing and the box sections at the bottom of the upper half on opposite sides, which assure rigidity in both the vertical and the horizontal directions.

It is thus well known in automotive engineering circles that in order to prevent roughness in operation, combustion must be so controlled that the rate of pressure rise is not excessive, and the crankcase or engine block must be so designed that it offers great resistance to transverse deflection. Any theory intended to explain the nature of roughness may therefore seem to come rather late to be of much help to engineers, but I feel that if we understand the nature of the phenomenon better we may learn of improved ways of dealing with it. For instance, it is always desirable to achieve a certain result with a minimum amount of material, and if we can throw some more light on the frequencies, amplitudes and directions of the impulses that cause the roughness, it may be of help in working out the most economical designs that are satisfactory from the standpoint of smoothness of operation.

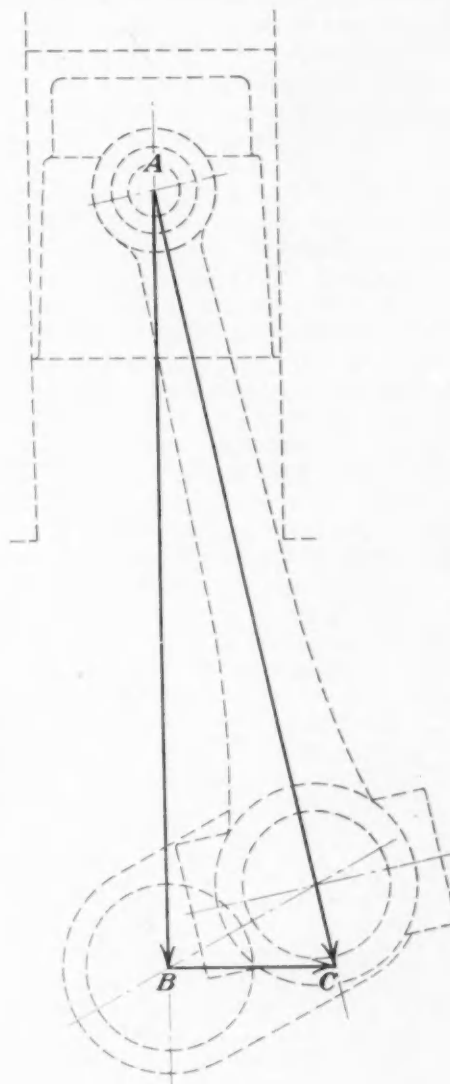


Fig. 2—Vertical and Horizontal Components of Pressure Transmitted by Connecting Rod

There are evidently two exciting forces that may result in rough operation, namely, gas pressures and inertia forces. The gaseous pressure is exerted in the form of impulses of short duration, and since these impulses are transmitted through the piston, connecting rod, and crankshaft to the main bearings in the crankcase, it might be expected that the crankcase would deflect downwardly under the impulse. However, it must not be overlooked in this connection that when the gaseous explosion deals a blow to the piston, it simultaneously deals an equal blow in the opposite direction to the cylinder head. In most modern engines the cylinders and the crankcase are in a single casting, and while the explosion therefore may tend to "stretch" the cylinder, as it were, it cannot elastically deflect the block in the vertical direction. In a few engines "through" bolts are used which tie the engine block, cylinder head and main-bearing caps together, and the entire stress due to the explosion is then taken on these bolts.

However, the entire force of the explosion is not spent in top dead center. In fact, in a high-speed engine the gas pressure reaches its maximum value only some time after the crank has passed the dead center position and there is a considerable pressure in the cylinder throughout the power

stroke. At any point of the cycle except the dead-center points, the angularity of the connecting rod produces a transverse or horizontal component. This is illustrated in Fig. 2, where the heavy vertical line *AB* represents the gas pressure on the piston. Because of a sort of wedge action due to the angularity of the connecting rod, this produces a greater force *AC* along the rod and a horizontal force *BC* on the main bearing. In an engine turning right-handedly this transverse force is toward the right during the power stroke and toward the left during the compression stroke. At the beginning and the end of the stroke the connecting rod is in line with the crank, and there is then no transverse force. The transverse force on the main bearings is counteracted by a similar force exerted by the piston against the cylinder wall, but since this force is separated from that on the main bearings by a distance which has an average value equal to the length of the connecting rod, it does not neutralize the effect of the transverse forces on the main bearings.

In Fig. 3 is shown a curve of the variation of the horizontal forces on the main bearings due to the gas pressure alone. This was drawn from the actual indicator diagram shown in Fig. 4, which indicates a maximum combustion pressure of about 375 lb. per sq. in. The transverse force on the main bearings for any crank position is easily found by first determining the gas pressure on the piston for that crank position and the angle which the connecting rod makes with

the cylinder axis, and then multiplying the gas pressure by the tangent of the angle. In this way points were plotted in the diagram of Fig. 3 and then a smooth curve drawn through these points. During the compression stroke, since the connecting rod is then inclined to the cylinder axis in the opposite direction, the transverse force on the main bearings is toward the left, while during the power stroke it is toward the right.

The oppositely directed horizontal impulses on the main bearings or on the crankcase, represented by Fig. 3, are repeated for each cylinder once per cycle, and in the different cylinders they follow one another in the firing order. Between succeeding impulses in each cylinder there is an interval corresponding to two piston strokes (exhaust and intake) when there is no gas pressure worth considering. Each cylinder therefore produces a succession of irregular transverse impulses on the crankcase through the main crankshaft bearings adjacent to its crank throw. According to Fourier, any irregular periodic force of this kind may be accurately represented by a constant mean force and a series of sine-wave forces of different frequencies. The sine-wave force of lowest frequency, known as the first harmonic, has the same frequency as the irregular periodic force, the fundamental. Then there are also sine-wave forces of two, three, four, five, etc., times that frequency, known as the second, third, fourth, fifth, etc., harmonics. If the fundamental is symmetrical to both sides of the base line, so that the mean value is zero, then there are only even-numbered harmonics. This, however, does not apply to the transverse force on the crankcase due to gas pressure, which is decidedly unsymmetrical with respect to the base line, as may be seen from Fig. 3.

The constant mean value of the horizontal impulses can be easily found by planimetry the areas bounded by the fundamental curve. This force, of course, cannot cause any vibration and is of interest in this connection only because it must be known before the values of the harmonics can be determined. The amplitudes and the phase angles of the different harmonics are determined from the fundamental by what is known as harmonic analysis or wave analysis.

Since explosions do not occur simultaneously in all of the cylinders, similar harmonics of the fundamental due to combustion in the different cylinders are out of phase. It is evident, for instance, that the first harmonics of the transverse forces could be all in phase only if explosions occurred in all of the cylinders simultaneously. As a rule the explosions are evenly spaced, and in a six-cylinder engine, for example, the first harmonics of the transverse impulses due to the different cylinders differ in phase by one-sixth of a cycle. This means that when the horizontal forces on the crankcase due to some of the cylinders are toward the right, those due to other cylinders are toward the left, and the sum total is nil.

It may be well here to explain a matter connected with the subject of frequencies in some detail. Ordinarily the time of a complete cycle of the basic periodic force is considered the period of the phenomenon, and the inverse of this figure is the frequency. In the case of internal combustion engine phenomena, however, it is more convenient to consider the time of one crankshaft revolution as the basic period and the number of revolutions per minute as the basic frequency. If we call this frequency N , then the frequency of engine cycles in a four-stroke engine is evidently $N/2$, since there is only one cycle every two revolutions. The first harmonic of the periodic force also has a frequency of $N/2$, the second harmonic of same has a frequency of N , and the third harmonic of $3N/2$, and so on. This explains what to

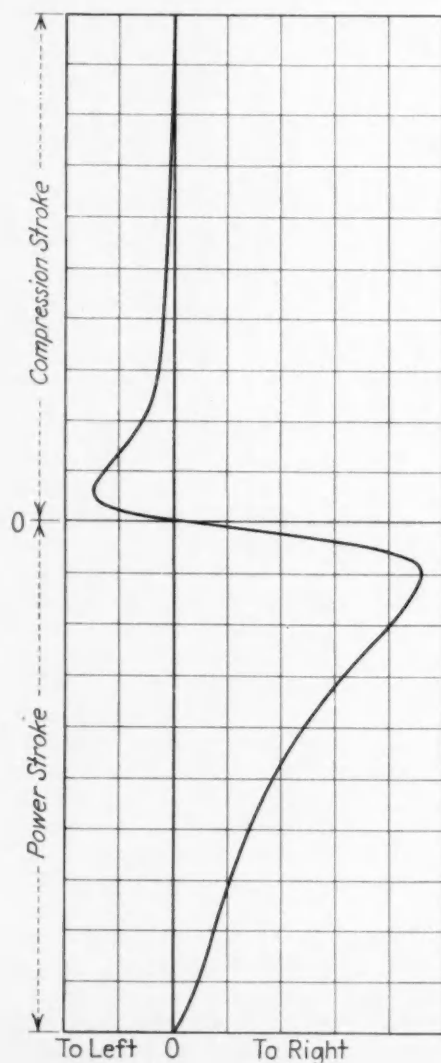


Fig. 3—Diagram of Horizontal Force on Crankcase Due to Gas Pressure During Compression and Expansion Strokes

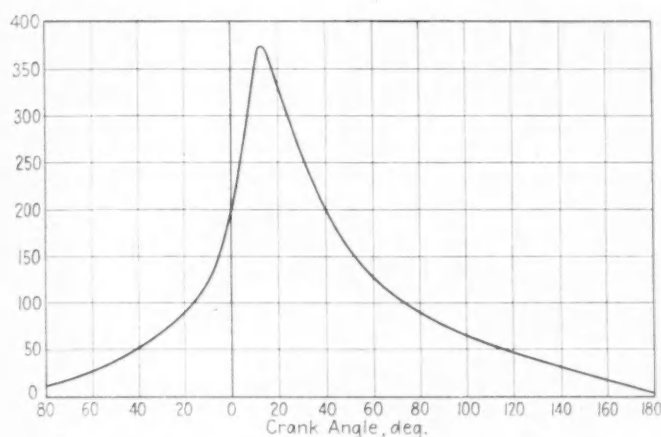


Fig. 4—Pressure-Time Diagram on Which Fig. 3 Is Based

some at least must have been rather puzzling, that harmonics with fractional designations, such as the $4\frac{1}{2}$ harmonic, are sometimes spoken of. There is no reference to such fractional harmonics in the Fourier theory. The explanation is that these designations are based, not on the period of the engine cycle, but on the period of the engine revolution, which is only one half a cycle, so that what is referred to as the $4\frac{1}{2}$ harmonic is really the ninth harmonic when the terminology explained in the foregoing is used.

It was shown in the foregoing that in a multi-cylinder engine the first harmonics of the transverse forces on the crankshaft due to the different cylinders differ in phase. These phase differences for a conventional six-cylinder engine are investigated in Fig. 5. The vector diagrams under the diagram of the six-throw crankshaft indicate the phases of the various harmonics. The top row, for instance, shows the phase relations of the first harmonics due to all six cylinders. These phases, as already stated, are uniformly distributed over the whole cycle. Thus the phase of the first harmonic due to cylinder No. 1 is in direct opposition to that from No. 6, those from Nos. 2 and 5 are in direct opposition, as are also those from Nos. 3 and 4. Conditions are a good deal the same for the second, third, fourth and fifth harmonics having frequencies N , $3N/2$, $2N$, and $5N/2$, and it is only when we come to the sixth harmonic, having a frequency of $3N$, that we find the harmonics due to all six cylinders in phase.

These harmonics, of a frequency equal to three times the crankshaft speed, tend to cause the crankcase to vibrate in the horizontal plane. This tendency exists at all speeds, that is, for all frequencies of the exciting forces, but the vibration can assume serious amplitudes only when the harmonics are in resonance with the crankcase as a vibrating body. The frequency of the crankcase is fixed, but the frequencies of all harmonics go up and down with the engine speed.

A body of the form of an engine crankcase is comparatively stiff in any case, and its natural frequency of transverse vibration therefore is quite high, but if the engine is speeded up the harmonics of frequency $3N$ may come into resonance with the crankcase, and the operation of the engine will then be rough. However, if the crankcase should be sufficiently stiff and its frequency of transverse vibration sufficiently high so that there can be no resonance between it and the harmonic of frequency $3N$, then there may be resonance between it and the harmonic of frequency $6N$ or $9N$, which are also in phase for all of the six cylinders. These

harmonics, however, are generally of much smaller amplitude, hence the vibration caused by them would be less severe.

Undoubtedly all of you have had experience with car engines which run very smoothly up to a certain speed, but as soon as you exceed 55, 60 or 65 m.p.h. the operation becomes noticeably rough. This is due to resonance between some harmonic of the gas pressure force or the inertia force and the engine block or crankcase.

In addition to the gas pressure force we have the inertia force due to the reciprocating masses in an engine as a possible cause of transverse vibration. In this case it is not even necessary to resort to wave analysis to prove the presence of harmonics of increasing frequency. In integrating the differential equation for piston motion we generally resort to a short cut which leads to a solution giving an expression for the momentary piston position consisting of two terms, the first term involving the cosine of the crank angle and the second term the cosine of twice the crank angle. The cosine of an angle is equal to the sine of its complement, for which reason curves of sines and cosines plotted on a base representing the circumference of a circle of unit radius are of exactly similar form. However, a strictly correct solution of the equation for piston position can be arrived at only by expanding the differential coefficient of the equation into a series and then integrating. This leads to a series of an infinite number of terms, successive terms of which involve the cosines of successive multiples of the crank angle. The expression for the inertia force also involves the cosines of these successive multiples of the crank angle.

There is one difference between the gas pressure forces and the inertia forces as regards their ability to cause vibration of the engine block. It was pointed out in the foregoing that since the gaseous pressure is exerted equally upwardly against the cylinder heads and downwardly against the pistons and through them against the crankshaft main bearings, this pressure cannot produce any transverse vibration of the cylinder block in the vertical direction (in a vertical engine). This does not apply to the inertia forces, however. These forces act on the block in one direction only at any one time, either upward or downward, since they originate in the free recipro-

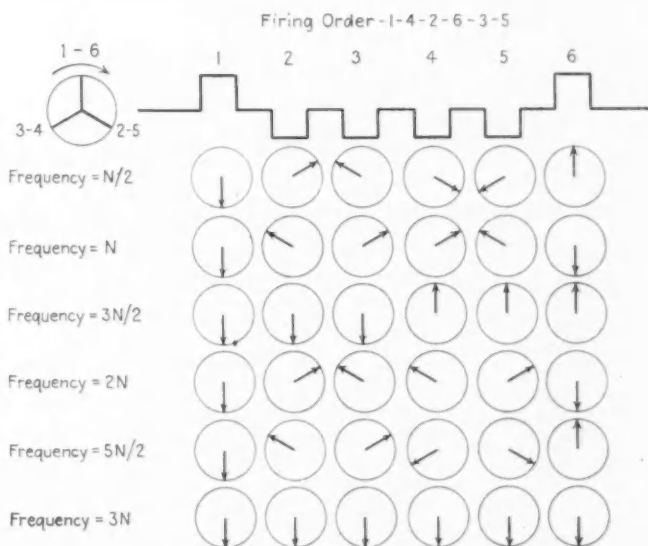


Fig. 5—Phase Relations of Various Harmonics of Horizontal Force Due To Gas Pressure in a Conventional Six-Cylinder Engine

cating mass. The inertia forces consist of a series of harmonics of frequencies N , $2N$, $3N$, etc. They also are transmitted by the connecting rods, and as a result horizontal components are created which tend to vibrate the crankcase horizontally, whereas the inertia forces themselves tend to vibrate it vertically. It will also be noted that the harmonics of the inertia force have the same frequencies as the harmonics of the gas-pressure force, except that there is no inertia-force harmonic of frequency $N/2$, the lowest frequency of the gas-pressure harmonic.

It is evident that the crankcase can vibrate transversely in any direction. The conventional crankcase is stiffest in the vertical direction, because of its two rather deep and substantially vertical side walls, and most flexible in the horizontal direction. Since the vibrating mass in transverse vibration would be substantially the same whether the case vibrated vertically or horizontally, it follows that the natural frequency of vibration of the case is lowest for vibration in the horizontal direction, and as the speed of the engine is increased, synchronous vibration is most likely to occur first in the horizontal direction. It must not be assumed, however, that the vibration must be either horizontal or vertical; the crankcase can vibrate transversely in any direction, and if the natural frequency of transverse vibration is, say, two or three times as great in the vertical as in the horizontal plane, then the crankcase can vibrate at any intermediate frequency in inclined planes between the horizontal and vertical. This undoubtedly explains that roughness is not confined to any narrow band of frequencies, like torsional vibration of the crankshaft, but covers a very wide range of frequencies beyond a certain minimum. In other words, if your engine has gotten into a zone of roughness you cannot, as a rule, get out of it by increasing the speed still further, but only by reducing the speed by closing the throttle.

In the case of a vibrating body of simple geometrical form it is usually possible to calculate the critical speed, that is, the rate at which impulses must be imparted to the body to produce resonant vibration. Even in the case of such a rather complicated assembly as a crankshaft with its attached parts, the calculation for the critical speeds of torsional vibration can be carried through with a considerable degree of accuracy. This calculation usually involves three steps. First the elastic vibrating body must be "reduced" to a simple geometrical form of the same stiffness. In the case of a crankshaft analyzed for torsional-vibration critical speeds, this resolves itself into finding the length of a plain cylindrical

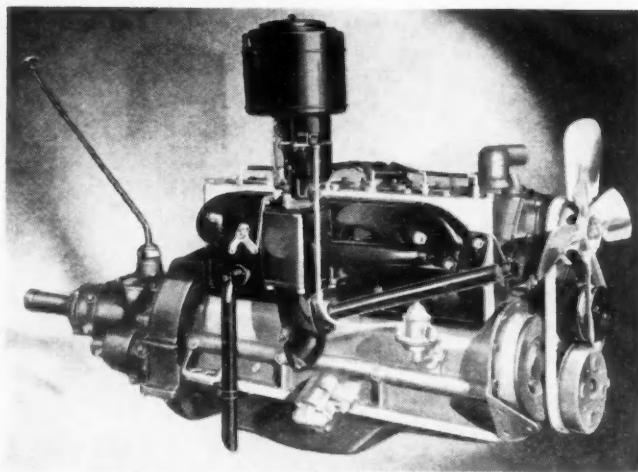


Fig. 7—Pontiac Engine with Bottom Flange and Heavy Oil Duct Running Lengthwise of Crankcase

shaft of the same diameter as the main bearings of the crankshaft which will deflect angularly through the same arc of circumference as the crankshaft when subjected to the same moment. The torsional stiffness of this plain shaft is readily calculated, and that of the crankshaft is the same.

The next step in the case of a torsional-vibration problem is to determine the moments of inertia of all of the moving parts of the assembly around the crankshaft axis. In a problem of transverse vibration the corresponding operation would consist in determining the mass which must be concentrated at the point along the length of the vibrating body where the amplitude of vibration is a maximum, to have the same amount of kinetic energy stored up when the vibrating body passes through the neutral position as the distributed mass of the actual vibrating body. After these two factors—the stiffness and the equivalent mass—have been determined, the frequency of natural vibration of the body can be readily found by means of the equation

$$n = \frac{30}{\pi} \sqrt{\frac{Rg}{W}}$$

where n is the number of cycles per minute; R , the flexibility of the vibrating body in lb. per in. of deflection; g , the constant of gravity in inch units (386 in. per sec. per sec.); and W , the equivalent vibrating mass concentrated at the point of maximum amplitude.

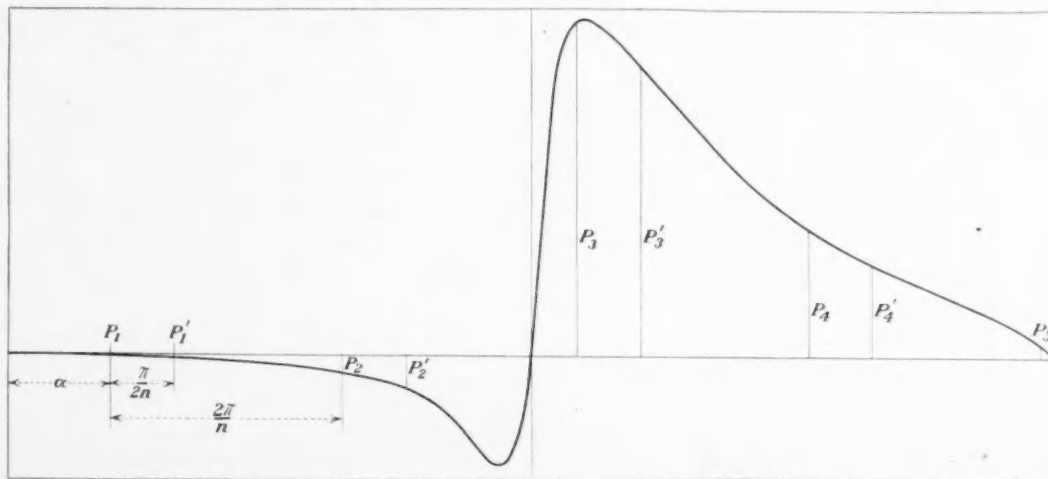


Fig. 6—Wave Analysis Diagram

With a complicated form like that of a crankcase it is quite impossible to calculate either the stiffness or the equivalent mass, and it is therefore impossible to determine the natural frequency from the dimensions and weight of the crankcase and its associated parts. Various instruments, generally referred to as vibrographs, are available for use in determining the actual frequencies of vibration of a body in all three planes.

In connection with a study of engine vibration the subject of harmonic analysis is of some interest. This, as already pointed out, consists in determining the amplitudes or maximum ordinates of the various harmonics composing a periodic curve of more or less irregular form, together with the phase angles of the individual harmonics. The harmonics, as stated, are double sine curves, and it is one of the properties of such a double-sine curve that if ordinates are erected on the base line at any two points separated by a distance $\pi/2$ along the base line (the length of the whole base line being 2π), the maximum ordinate of the curve is equal to the square root of the sum of the squares of these two ordinates.

We now take a periodic curve similar to that shown in Fig. 3, representing the transverse horizontal force on the crankcase due to the gas-pressure impulse in one cylinder. This is only one-half of the complete cycle, but since there are no gas pressures of any consequence during the exhaust and inlet stroke, that portion of the cycle representing these two strokes is here omitted. In Fig. 6, α represents any angular distance from the beginning of the cycle, which we have here taken to coincide with the beginning of the compression stroke. Starting at point α on the base line, the cycle is divided into nine equal parts and ordinates $P_1, P_2, P_3 \dots P_9$ are erected at the division points. It is obvious, of course, that since the area of the right-hand half of Fig. 6, which represents the power stroke, is much greater than that of the left-hand half, there is a constant mean force acting in the upward direction in Fig. 6. The value of this is first determined by measuring the areas (by planimeter, for instance), taking the difference between them, dividing by the length of the base line, and multiplying by the scale of ordinates. We will assume that this operation has been carried through and that the constant has been found to have a value C .

If now we add up all of the nine ordinates P algebraically (several of them, of course, are equal to zero in this case) and then divide the sum by 9, the quotient is equal to the sum of the constant C and the ordinates at point α of the ninth, eighteenth, twenty-seventh, etc., harmonics. The series of harmonics is infinite, but the harmonics of large order are very small in magnitude and it is therefore permissible to neglect all those above a certain order. It is usually only the first two or three harmonics that are of any consequence, but to obtain these with a fair degree of accuracy we must obtain approximate values for a good many more.

Referring to Fig. 6,

$$\frac{P_1 + P_2 + P_3 \dots + P_9}{9} = C + H_9 \sin 9(\phi + \alpha) + H_{18} \sin 18(\phi + \beta) + \dots$$

where H_9 is the maximum ordinate of the ninth harmonic; H_{18} the maximum ordinate of the eighteenth harmonic; ϕ the crank angle measured from the beginning of the compression stroke; α , the phase difference between the ninth harmonic and the fundamental; β , the phase difference between the eighteenth harmonic and the fundamental.

As stated above, because of the small amplitude of the harmonics of higher order, it is permissible to neglect them, and we will neglect all harmonics above the ninth. Also, we will replace the left hand side of the above equation, that is, the algebraic mean of the nine ordinates, by M . We then have

$$M = C + H_9 \sin 9(\phi + \alpha)$$

and if we let the crank angle ϕ be zero,

$$M = C + H_9 \sin 9\alpha$$

Next we erect a similar set of nine ordinates to the fundamental curve, but instead of starting at the zero point of the fundamental, we start at a point $\pi/18$ farther on (one quadrant on the base of the ninth harmonic farther on). We then get ordinates $P'_1, P'_2, P'_3 \dots P'_9$. Let the algebraic mean of these ordinates be M' . We then have

$$\begin{aligned} M' &= C + H_9 \sin 9(\alpha + \pi/18) \\ &= C + H_9 \sin (9\alpha + \pi/2) \\ &= C + H_9 (-\cos 9\alpha) \\ &= C - H_9 \cos 9\alpha \end{aligned}$$

Therefore

$$H_9 \sin 9\alpha = M - C$$

$$H_9 \cos 9\alpha = C - M'$$

and since $\sin^2 + \cos^2 = 1$

$$H_9 = \sqrt{(M - C)^2 + (C - M')^2}$$

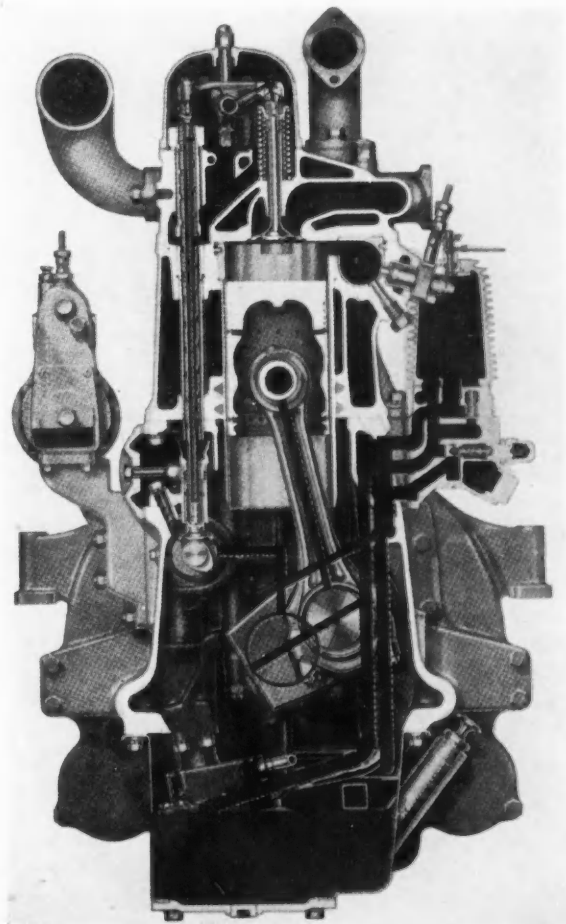


Fig. 8—Transverse Section of Hercules Diesel Engine Showing Reinforcement at Bottom of Crankcase

Since M , C and C' are obtained from measurements on the fundamental curve, we can readily determine the maximum ordinate H_9 of the ninth harmonic.

To obtain the phase angle we must remember that for an angle $\phi = 0$ on the base line, that is, at the beginning of the fundamental curve, the ordinate to the ninth harmonics is

$$H_9 \sin 9\alpha = M - C$$

from which it follows that

$$\begin{aligned} \sin 9\alpha &= \frac{M - C}{H_9} \\ &= \frac{M - C}{\sqrt{(M - C)^2 + (C - M')^2}} \end{aligned}$$

The phase angle 9α may have any value between 0 and 2π or 360 deg. It is in the first quadrant if both $M - C$ and $C - M'$ are positive; in the second quadrant if $M - C$ is positive, $C - M'$ negative; in the third quadrant if both are negative, and in the fourth quadrant if $M - C$ is negative and $C - M'$ positive.

In exactly the same way we may find expressions for the maximum ordinates and the sines of the phase angles of the eighth, seventh and sixth harmonics, in each case disregarding the harmonics which are even multiples of the particular harmonic under investigation. When we come to the fifth harmonic, however, it is advisable to take account of the influence of the tenth harmonic on the result. We therefore first determine the values for the tenth harmonic in the same way we determined those for the ninth in the foregoing.

We then have

$$M_5 = C + H_5 \sin 5(\phi + \lambda) + H_{10} \sin 10(\phi + \delta)$$

Since the values of H_{10} and δ are known, we can readily calculate the value of the last term on the right hand side for any given value of ϕ . For $\phi = 0$ this expression becomes $H_{10} \sin \delta$, which we will represent by Q .

Then

$$M = C + H_5 \sin 5\lambda + Q$$

$$H_5 \sin 5\lambda = M - C - Q$$

also,

$$M' = C + H_5 \sin 5(\lambda + \pi/10) + H_{10} \sin 10(\delta + \pi/20)$$

$$M' = C - H_5 \cos 5\lambda - H_{10} \cos 10\delta$$

Let $H_{10} \cos 10\delta = R$

Then

$$M' = C - H_5 \cos 5\lambda - R$$

$$H_5 \cos 5\lambda = C - R - M'$$

$$H_5 = \sqrt{(M - C - Q)^2 + (C - R - M')^2}$$

and

$$\sin 5\lambda = \frac{M - C - Q}{\sqrt{(M - C - Q)^2 + (C - R - M')^2}}$$

All of the factors of the right-hand side of the last two equations are known, hence the maximum ordinate and the phase angle of the fifth harmonic can be calculated. In the same way the fourth, third, second and first harmonics are determined.

Since "roughness" consists in transverse synchronous vibration of the crankcase or engine block, the problem that confronts the designer is to so design the case that it offers maximum resistance to such vibration, or to lateral deflection, with a given amount of material. Gas-pressure forces tend to cause it to vibrate horizontally in a vertical engine. These forces act on the crankcase at the crankshaft bearings and therefore the best provision against such vibrating forces is to stiffen the case horizontally in the plane of the crankshaft axis. For a long time there has been a tendency to make crankcases very deep, so as to obtain great vertical stiffness, and it is a question whether this has not seriously reduced the horizontal stiffness, because it tends to give flat side walls. Also, with a view to increasing the vertical stiffness, the parting plane between the upper and lower sections of the crankcase has been brought several inches below the crankshaft axis. This brings the bottom flange of the upper section, which usually contributes most to the horizontal stiffness, out of line with the points of attack of the horizontal vibrating forces. In a good many of the more recent designs this has been remedied by extending a heavy rib along the side of the crankcase wall at the level of the crankshaft axis. This rib, in conjunction with the bottom flange and the side wall of the crankcase connecting them, forms a channel section which affords considerable vertical stiffness. It is rendered even more effective as a beam by connecting the bottom flange and the rib by a number of vertical ribs suitably spaced.

Fig. 7 and Fig. 8 each show an example of recent design in which particular attention has been given to lateral stiffness of the crankcase.

In some cases, as in the Waukesha engine shown in Fig. 1, box sections are used to give adequate stiffness to the crankshaft in both the horizontal and the vertical directions. A high-speed Diesel built by Burmeister & Wain of Copenhagen for railcar service (Fig. 9) also has such a box section at the bottom of the crankcase upper section.

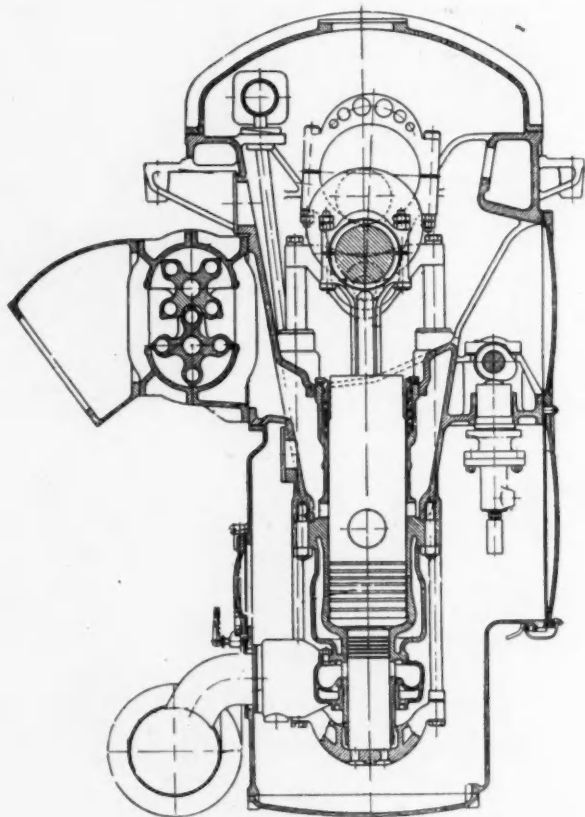


Fig. 9—Section of Burmeister & Wain Diesel Engine with Box Section at Bottom of Crankcase

Service Problems from Manufacturer and Operator Viewpoints

By O. M. Brede

General Motors Truck Co.

THIS paper centers attention on operating cost control, which is stated to be the most difficult of all the service problems to solve. Until the scientific as well as the mechanical phase of maintenance is recognized, until repairs are made from facts rather than guesswork, and until the question: Why this failure? takes precedence over "repairing," the author states that this serious service problem cannot be solved.

The danger of confusing cost records with maintenance factual information is emphasized. Only a maintenance history, a chronological array of facts embracing repairs, repeat work, road failures and the like, can answer the question: What caused this cost?

Preventive maintenance has as its fundamentals regularity, uniformity and thoroughness, and can be applied to any single vehicle or fleet regardless of whether self-maintenance or service-station maintenance is practised.

Numerous illustrations of maintenance troubles and forms for record purposes are given and commented upon. In conclusion, the author advocates the promulgation by the industry of a preventive-maintenance system.

CONSIDERED from both the manufacturer's and the operator's viewpoint, the subject is indeed broad and its divisions manifold. The problems begin on a drafting board in the engineering department of the manufacturer and end only when the vehicle has traveled its last mile. These problems have grown proportionately with the evolution of the automotive industry. From a modest beginning—not so many years ago—statistics record approximately 3½ million motor trucks registered in the United States at the

present time. Such growth entails many service problems, both external and internal. They involve many factors which, if they are to be solved, demand the efficient coordination of many relationships working upon sound principles.

The physical factors of service problems have attained a fair degree of perfection. It is possible to find mechanics in most every hamlet, town or city of the country—of varied abilities, of course—but, generally speaking, good automotive mechanics can be obtained almost anywhere—as can supervisors—capable of directing the efforts of others, and qualified through years of experience to differentiate between proper or inferior workmanship. This makes the repair problem—viewed and treated strictly as a repair—relatively simple.

Likewise: parts availability, service-station facilities, tool development, equipment selection, and the like, are factors which have progressed to a point where they do not in my opinion constitute serious service problems at present. Truck manufacturers, service-station builders, tool and equipment manufacturers and fleet-operating personnel are working individually and collectively on these physical factors. With such an array of thinking, there is little doubt that progress will keep pace with necessity.

Turning to the management, or administrative side of service, however, there is one very serious problem which does not reflect the improvement, nationally, that its importance deserves; and because it would not be possible to discuss several problems other than very sketchily in the space available, I have chosen to confine my subject to the problem of Operating-Cost Control.

In my opinion, this is the most difficult of all the service problems the maintenance industry has been called upon to solve. More difficult than others, I contend, because of the new fields the motor truck has conquered. Instead of being safely housed in the operator's place of business each night, it may—like an ocean vessel—be many miles away from its home base. Instead of delivering a load and returning at leisure, split-minute schedules are maintained. Perishable loads are required to meet crack trains; castings must be delivered on time to avoid paralyzing production in large factories; department-store-delivery promises must be kept in order to retain customer good will; and so on.

The very reason for the remarkable registration increase, namely, the adapting of the motor truck to new tasks, has complicated the problem of operating-cost control; and until

[This paper was presented at the Regional Transportation and Maintenance Meeting of the Society, Newark, N. J., Oct. 30, 1935.]

the scientific as well as the mechanical phase of maintenance is recognized; until repairs are made by facts rather than by guess; until the question: "Why This Failure?" takes precedence over "Just Repairing", this serious service problem cannot be solved.

I recognize, of course, that individuals in various sections of the country have made and are making remarkable progress in lowering operating costs in their respective operations. This is especially true in large corporation-controlled fleets where shrewd administrations—fortified with accurate cost information—have recognized the absolute necessity of economical transportation. Even these large operators, however, are in some instances guilty of confusing cost records with maintenance factual information.

True, accounting will reveal costs; but only a maintenance history—a chronological array of facts embracing repairs, repeat work, road failures, and so on—can answer the question: "What Caused This Cost?"

Our analyses of fleets—both large and small, concentrated and spread, a number of which must depend upon the cost of delivery and the service rendered for their business existence—have proved that many operators are, as of today, utterly at sea with respect to knowing the facts about their operation; and because of this, they cannot effect permanent reduction in operating costs.

Vehicle difficulties in these operations are usually measured only in the mechanical sense; the trouble and cost experienced in the repair rather than the cause of the difficulty. Also, insufficient value is placed upon vehicle earning capacity and the subsequent loss of revenue for "out-of-service time." This factor alone, when measured in dollars per hour, is in some instances as much as the entire maintenance bill.

The amount of money that many operators—large and small—spend every year to support a "guessing contest", is nothing short of tragic. The "guessing" we refer to is often disguised under that \$7 word—"diagnosis." The guess may be right the first time; but, too often—from the standpoint of an operator's pocketbook—it is right the last time. Again, out-of-service time, together with the wasted efforts of mechanics, the substitution of parts and the driver's inactivity, are buried in a mass of figures reflected somewhere on the balance sheet as "operating costs."

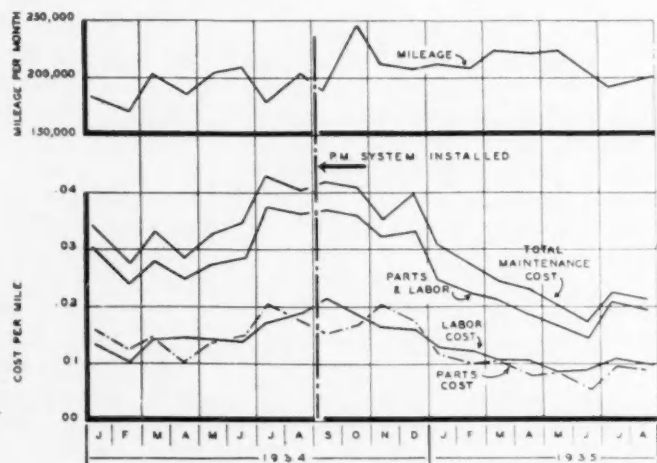


Fig. 1—Operating Costs—Before and After Preventive Maintenance

A decided downward trend is noted in all items contributing to total maintenance costs. The letters beneath the curves denote the months of the year.

If the thought could be thoroughly impressed on all personnel connected with fleet operations that repairing in fleets is the very last resort of maintenance—that every other means should be utilized before having recourse to repairing and that the efficiency of the maintenance of the fleet is inversely proportional to the amount of repairing undertaken—great strides would be made in the development of fleet maintenance. True, there comes a time when repairing is inevitable; but the maximum effort should be concentrated on postponing this time to the very farthest possible date. In other words, the aim and objective of maintenance should be primarily to *prevent* repairs. And thus we find a phrase which is slowly but surely being heard throughout the industry; namely, *Preventive Maintenance*.

Preventive maintenance has as its fundamentals *regularity, uniformity and thoroughness*, and may be applied to any single vehicle or fleet, regardless of whether self maintenance or service-station maintenance is practised. To illustrate

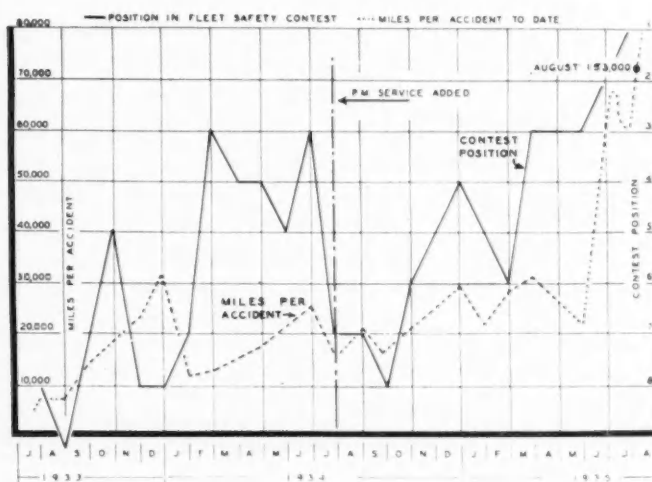


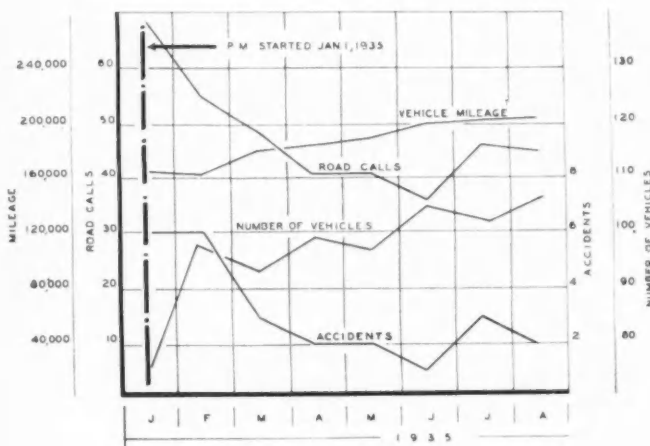
Fig. 2—Accident History

The number of miles between accidents took an upward turn soon after the installation of a preventive maintenance system. In August, 1935, this fleet traveled 153,000 miles without an accident. The letters beneath the curves denote the months of the year.

graphically what can be accomplished, we have charted three fleets: Fig. 1, (fleet A) showing maintenance cost per mile; Fig. 2, (fleet B) showing accident history; and Fig. 3, (fleet C) showing road failures and accidents.

Let us review Fig. 1. This chart covers a period of 20 months. Preventive maintenance was installed during the eighth month. It will be noted that while mileage traveled remained fairly constant at approximately 200,000 miles per month as shown by the top curve, maintenance costs showed a decided drop immediately following the installation of preventive maintenance in August of 1934. The total maintenance costs illustrated which had been steadily increasing and which were over 4 cents per mile in July, 1934, were reduced to slightly over 2 cents per mile by August, 1935, and the curve is continuing downward.

The total parts and labor costs represented in Fig. 1 were reduced from 3.8 cents per mile to 1.9 cents per mile. Labor costs, as indicated, were reduced from slightly over 2 cents per mile to approximately 1 cent per mile, while an almost identical improvement in parts costs is shown by the parts-cost curve. Comparing total maintenance costs for the month of August, 1934, with August, 1935, there is a saving of



approximately 2 cents per mile or \$4,000 per month, since mileage has remained constant at approximately 200,000 miles per month.

Fig. 2 shows the accident history of a fleet of approximately 200 trucks operating in a large eastern city. This particular city presents a safety award each month to the fleet having the best safety record. The solid-line curve shows the position of the fleet in the contest and it will be noted that the fleet was in ninth position in August, 1933. While some improvement is reflected during succeeding months, this improvement is extremely erratic until shortly after the installation of preventive-maintenance service in July, 1934. From that time on improvement was rapid and in August, 1935, the fleet won the monthly safety award. Reference to the dash-line curve, which illustrates miles traveled per accident, shows that in July, 1933, accidents were so numerous that it was difficult to determine the exact mileage between them. Steady improvement followed the installation of preventive-maintenance service, however, and in August, 1935, this fleet had the best record in the city; namely, 153,000 miles without an accident.

Fig. 3 illustrates an operation in the city of Philadelphia. Preventive maintenance was installed in January, 1935. At that time there were approximately 75 trucks in the fleet; but, as of Aug. 1, the fleet had grown to 108 trucks. As indicated by the "road-calls" curve, the equipment averaged approximately one road failure per truck per month at the time preventive maintenance was begun. Although mileage has increased over 25 per cent and the fleet has been increased almost 50 per cent, road failures show a decided downward trend. A low was reached in June, when 38 road delays were charted and, although the experience in July was not satisfactory, the curve resumed a downward trend in August. At the time preventive maintenance was installed, six accidents per month or approximately one accident to each eleven vehicles occurred. This has been reduced to one accident for each fifty-four vehicles per month as of August, 1935. The figures shown in these charts are not to be construed as representing the ideal—either in maintenance cost per mile, miles traveled per accident, or miles per road failure—but to illustrate the immediate effect of preventive maintenance on various phases of fleet operation.

I mentioned previously that individuals have made and are

making remarkable progress in controlling and lowering operating costs, but there are thousands of operators whose costs are not controlled and who practice preventive maintenance only partially or not at all. To those operators who are still repairing by guess, rather than by fact, the following preventive-maintenance system is offered. It is not cumbersome nor intricate but, on the contrary, simple, easy to operate and one that records facts.

There are two phases of this preventive-maintenance system; one embracing the mechanical side of the service and the other, the accumulation of facts. In outlining the system, I will deal with these phases separately, referring first to the mechanical.

Four simple forms guide the mechanic through inspection, lubrication and repair. These forms, it must be understood, are guides only, subject to such revision as each specific operation warrants. The removal of items which do not affect certain vehicles, or the addition of special items which may not be included, is, of course, expected. These forms are identified as *A*, *B*, *C* and *D* Services, to be performed at various mileages. Exact periods are established by maintenance history and governed by the type of operation. Later, I will explain how mileage periods for the various services can be determined; but at this point let us assume that, after proper analysis, the system has been installed in a given fleet and periods for the various services have been established at *A*, 1000 miles; *B*, 5000 miles; *C*, 15,000 miles; and *D*, 30,000 miles.

The A preventive-maintenance service, shown in Fig. 4, is performed each 1000 miles; or, on low-mileage vehicles, each 30 days. This service includes, in addition to lubrication and definite items of inspection, several adjustments having a de-

FORM NO. G-60 (REVISED 10-6-66)

General Motors Truck Company PREVENTIVE MAINTENANCE WORK SHEET

MODELS T76 to T93 INCLUSIVE

"A" Service (Suitable at 1000 MI. Intervals)

OWNER _____	MODEL AND CHASSIS No. _____	DATE _____
ADDRESS _____	MILEAGE _____	R. A. No. _____

(V) O. K.

(X) ADJUSTMENTS MADE

(C) REPAIRS NEEDED

ITEMS OF ADJUSTMENT AND LUBRICATION

<p>1 Check and adjust spark plugs— Gas openings—(SI Draft Carburetor) #18 to #23 (Up Draft Carburetor) FILDS SHOULD BE REPLACED AT 1000 MI.</p> <p>2 Check and inspect distributor points— Point opening .018—.022 CONDENSER SHOULD BE REPLACED AT 1500 MI.</p> <p>3 Check ignition timing, oil distributor work— (all distributor gears ok)</p> <p>4 Adjust fan belts—(Not too tight)</p> <p>5 Tighten Water Pump/Gland Nut and Hose connections</p> <p>6 Check fuel pump strainer and bowl and carburetor strainer. FUEL PUMP SHOULD BE RECONDITIONED AT 1500 MILES</p> <p>7 Tighten cylinder head, manifold, governor and carburetor flange bolts</p> <p>8 Adjust valve clearance—(in overhead gear oil) #80 to #11—Overhead #8 to #10—215 Engine #12 to #16—321, 237, 340, 400, 430 Engine</p> <p>9 Adjust carburetor (set with vacuum gauge where possible)</p>	<p>10 Grease thoroughly (a) Fuel injectors or air/fuel lines (b) Oil or grease universal joints, check for looseness (c) Oil and inspect all brake linkages and pedal shaft (d) Oil governor, starter and Distributor, Fan, Choke, Carburetor Linkage, Hood Pulls, Door Hinges and Locks. (e) Grease Water Pump</p> <p>11 Check front wheel bearings and adjust if necessary and inspect steering knuckle pins for looseness WHEELS SHOULD BE REMOVED—BEARINGS CLEANED AND REPACKED WITH FRESH GREASE EACH 12,000 TO 15,000 MILES</p> <p>12 Adjust clutch pedal for clearance.</p> <p>13 Crank speedometer cable each 10,000 miles</p> <p>14 Tighten front spring U-bolts</p> <p>15 Tighten all wheel nuts</p> <p>16 Tighten axle shaft nuts or flange bolts WHEELS SHOULD BE REMOVED—BEARINGS CLEANED AND REPACKED WITH FRESH GREASE EACH 12,000 TO 15,000 MILES</p> <p>17 Inflate tires and align after wheeling: R.F.—No.; R.R.—No.; L.P.—No.; L.R.—No.</p> <p>18 Tube hydrocarbon cooling—add battery acid _____ and fill with distilled water</p>
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ITEMS OF INSPECTION

<p>19 AIR HYDRAULIC AND BOOSTER BRACKES— Tighten all fluid or air line connections—check line pressure (all) check master cylinder (fluid level (hydraulic) check booster cylinder alignment, examine hose, add 2 cc. oil each end booster cylinder every 3000 MI. (cold test oil-pressure test at least 30 degrees below base)</p> <p>20 Inspect universal joints for looseness</p> <p>21 Check rear axle lubricant level, add lubricant if necessary. Examine for oil leaks at joints or worn.</p> <p>22 Check transmission lubricant level, add lubricant if necessary. Examine for oil leaks-front and rear of transmission</p> <p>23 Check body or cab hold down bolts for looseness</p> <p>24 Inspect radius rods</p> <p>25 Inspect rear spring U-bolts and alignment clips</p> <p>26 Check play in steering wheel</p> <p>27 Check parking brake—adjustment not included</p> <p>28 Inspect drag link and steering arms for looseness and out-of-alignment</p> <p>29 Tighten starter switch and Ammeter Connections</p> <p>30 Check radiator oil level</p> <p>31 Check oil in all gasoline tanks</p>	<p>32 Check for grease leaks</p> <p>33 Inspect Radiator hold down bolts</p> <p>34 Inspect all filters: DILUXE—Inspect elements—if top is dirty replace filter element—Add operation 1-6-60 A. C. CYCLE REPLACE EVERY 5,000 MILES, add operation 1-6-60 PURBLATOR—clean element each 1,000 miles—Add operation 1-6-60</p> <p>35 Inspect condenser—connections—CLEAN EACH 5,000 MILES—ADD OPERATION 1-6-60</p> <p>36 Inspect air cleaner—Add operation 15-3620 for cleaning</p> <p>37 Check and Renew Oil Pressure Charging Rate</p> <p>38 Check Gas Gauge, Air Gauge, Frost Indicator, Horn</p> <p>39 Test all lights</p> <p>40 Fill radiator with water or check anti-freeze as necessary</p> <p>41 Check all doors, door latches and window regulators</p> <p>42 Check condition of fenders, running boards, splash guards, and top covering</p> <p>43 Road test vehicle—check brakes and speedometer, and semi-general conditions</p>
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REMARKS:

G-60(MP)

Fig. 4—"A" Preventive Maintenance Service

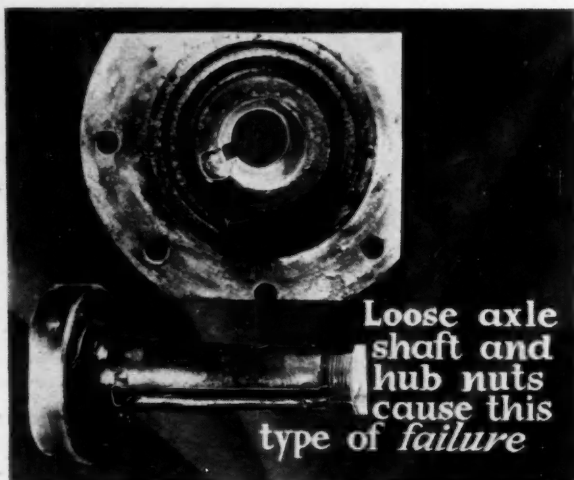


Fig. 5—Broken Axle Shaft and Worn Hub

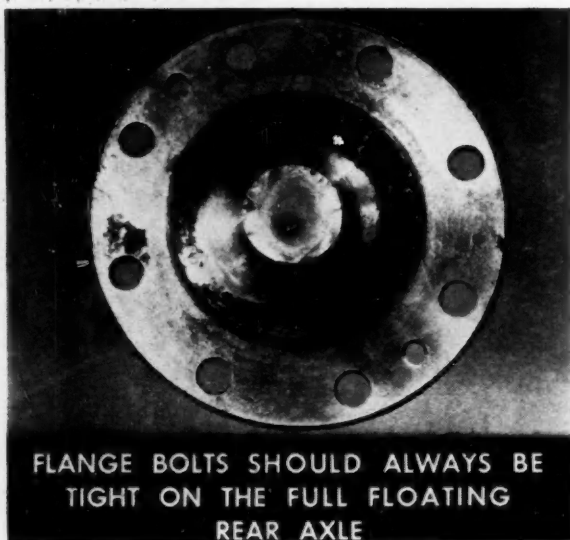


Fig. 6—Broken Axle Shaft at Flange End

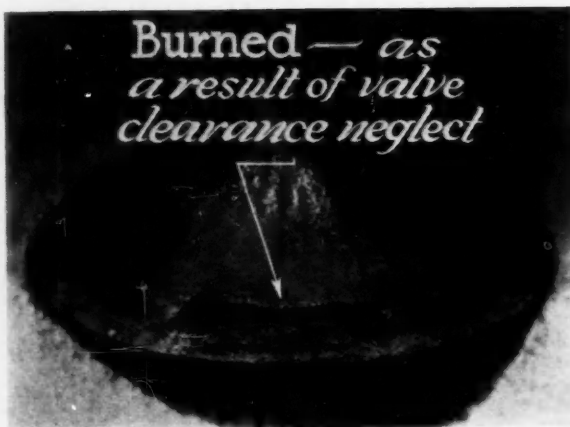


Fig. 7—Burned Valve

cided effect upon operating economy; for instance, spark-plug cleaning, tappet adjustment, front-wheel alignment, axle-shaft-flange tightening and others. I will not go into complete detail on each of these items; but, to illustrate our reasons for including definite items of adjustment in a service which is often confused with an inspection, I will show a few examples of what happens to various units of a vehicle when preventive maintenance is not in effect. These illustrations were obtained in a fleet operating five different makes of equipment, indicating that neglect plays no favorites.

Loose axle shaft and hub nuts cause the broken axle shaft and worn hub type of failure shown in Fig. 5. Its prevention involves only a few minutes each 1000 miles. Fig. 6 shows a broken axle shaft at the flange end. This is true of either semi- or full-floating-axle construction.

Correct wheel alignment will add thousands of miles to tire life. Insufficient attention is paid to the fact that failure to maintain accurate clearance between the valve rocker arm or valve lifters and valves affects power, economy and valve life. The burned valve shown in Fig. 7 was burned as a result of valve clearance neglect.

Fig. 8 shows hot and cold spark plugs. From an economy standpoint, we lay particular stress on the ignition system. Spark plugs must be cleaned and gaps correctly adjusted every 1000 miles. Unless this is done, actual tests prove that an engine loses approximately 10 per cent of its efficiency on both power and fuel. The cleaning and spacing of the plug is not the only essential. Regular inspections will indicate whether or not the correct plug is being used for the service in which the vehicle is employed. The color of the porcelain tells the story. It will vary from a black color—indicating a very cold plug—to an ashen white, indicating a very hot plug. With carburetor properly adjusted, the porcelain color on a truck-type spark-plug should be light chocolate.

A lubrication chart is shown in Fig. 9. A thorough chassis lubrication is absolutely essential each 1000 miles or each 30 days. A word about lubrication. One of the most prevalent misconceptions regarding lubrication is that it can be performed by inexperienced, or plainly speaking, "cheap" help. This is one of the most costly mistakes that any operator can make. Today's operating conditions demand several types of lubricants on the same vehicle. Any fleet may contain two or three types of universal joints, each of which requires lubricants adapted to its construction. The same lubricant will *not* produce the same results on both light-duty and heavy-duty equipment. Various types of axles and the conditions under which they operate determine the type of lubricants which should be used in a differential, and the same thing applies to the transmission and the engine. Therefore, the selection and application of the correct lubricant requires the best thinking in the shop. The oil and automotive industries spend huge sums annually, and assign their best engineering talent to the development of correct lubricants. Their recommendations must be followed in the interests of longer unit-life. There are a total of 43 items of inspection, adjustment and lubrication in the *A* preventive-maintenance service. For complete details, see Fig. 4.

The *B* Service, specified in Fig. 10, which I have associated with 5000 miles, is more thorough. It also includes the *A*-Service items. It includes, among other items, a steering-system inspection, which includes a thorough tightening and adjusting of the complete steering gear. Fig. 11 shows a steering worm and housing that has failed. Neglect of the steering system results in such failures.

Fig. 12 shows a crankcase ventilator. Failure to clean the crankcase ventilator at proper intervals results in piston rings sticking in grooves, in sludge collection, in sulphuric-acid attacks upon the metal, and the like.

Fig. 13 shows a brake drum and brake shoes that failed. Thousands of miles are added to brake life by frequent attention. Neglect makes necessary brake drum and, sometimes brake-shoe replacement, both of which are very costly. Although everyone knows that a battery requires care, its neglect continues to cause road failures. These and other items are given attention in the B Service. For complete details see Fig. 10.

The third service in the group, the C Service, shown in Fig. 14, which embraces both the A Service and the B Service is performed at advanced mileages and is identified here as the 15,000-mile service. It involves arbitrary replacement of minor parts which have served their economic usefulness and also includes the opening-up of certain concealed units which should be periodically examined. At the established mileage for the C Service, it will generally be found advisable to arbitrarily replace distributor points, distributor cap, condenser and ignition wires. These are shown in Fig. 15. A short vehicle-delay because of a failure of one of these parts probably would pay for several replacements.

A fuel pump is shown in Fig. 16. Fuel pumps are not receiving sufficient attention and many preventable failures result.

During the C Service, the entire cooling system, shown in Fig. 17, is thoroughly inspected, cleaned and refilled with fresh water. It includes replacement of hose and water-pump packing. Periodic attention to cooling systems prevents a condition such as illustrated.

Rear axle and transmission receive attention in the C Ser-

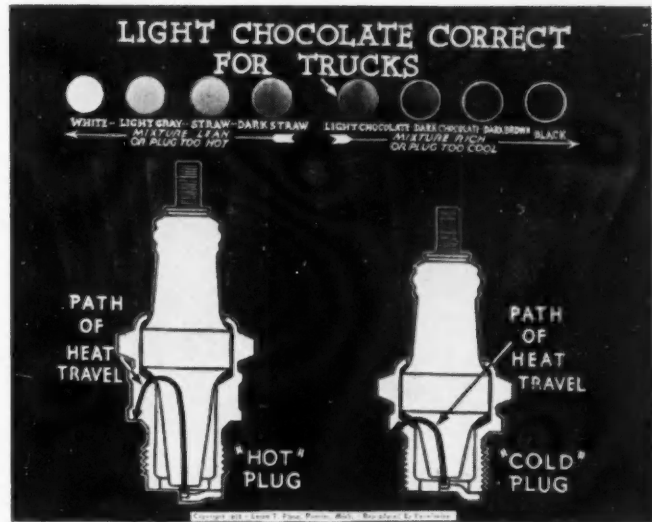


Fig. 8—Hot and Cold Spark-Plugs

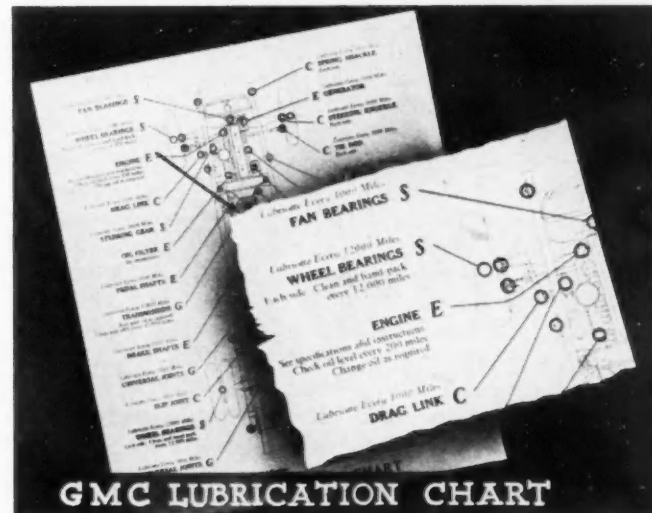


Fig. 9—Lubrication Chart

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General Motors Truck Company PREVENTIVE MAINTENANCE WORK SHEET MODELS T-16 to T-43 INCLUSIVE

"B" SERVICE

(INVESTIGATION OF VEHICLE AND RECORDS)

OWNER	MODEL and CHASSIS No.	DATE
ADDRESS	MILEAGE	R. A. No.
(✓) O. K.	(X) ADJUSTMENTS MADE	(O) REPAIRS NEEDED
1. SPARK PLUGS: Clean plugs—adjust gaps. Gap should be .015 in. (0.015 in. down draft, 0.015 in. up draft). Report on: Visual Inspection—Ignition—Spark Plug Too Hot	2. DISTRIBUTOR: Includes: Clean rotor and cap—Oil Dist. Wick—Clean Cam—Fill Grease Cup—Report on: Voltage at Points when open—V. (Should be same as battery). Voltage at points when closed—V. (Should be zero). Shalt—Cam—Automatic Advance—Primary Wiring—Ignition Switch—	3. COIL: Test Coil—Amperage draw should be 1.4 at approximately 15 m. p. h. Spark Gap—mm. Amperage draw—Amperes.
4. FUEL PUMP: Clean filter bowl and strainer—Test pump outlet pressure—1/4 in. T-16 to T-43, 4 in. T-46 to T-49—Test pump inlet vacuum—6 inches minimum—Tighten line connections and pump to case—Report on case gasket—	5. STARTER: Includes: Tighten cable connections at starter—Oil Starter—Report on: Commutator—Brushes—Drive—Switch—	6. GENERATOR: Includes: Adjusting charging rate if necessary—Adjusting fan belt—Tighten Ammeter connections—Oil generator—Report on: Commutator—Brushes—Ammeter charging rate—Amps. Cut-out Relay—
7. CRANKCASE VENTILATOR: Includes: Remove Clean and install—	8. OILING SYSTEM: Includes: Clean or replace oil Filter—Tighten oil line connections—Report on: Oil Pressure—No. Oil level in pan—Oil leaks—	9. COOLING SYSTEM: Includes: Tighten Water pump gland nut or grease pump—Tighten Hose connections—Tighten radiator hold down bolts—Tighten radiator Tie rods—Report on: Radiator—Hose—Anti-Freeze—Pump Packing—Pump Shaft—Fan—Fan Belts—
10. VALVES: Tighten head bolts—Each valve, each should be 200 mg. (200 - 211, 213 mg. 0.008 - 0.010, 221 to 4.0 mg. 0.012 - 0.014. Includes: Report on Springs—Clean overhead with air—Replace cover gasket if necessary—	11. CARBURETOR: Includes: Clean air cleaner—Tighten Manifold—Carburetor Flange—Exhaust pipe Flange—Governor Flange—Oil choke and Throttle linkage—Adjust Carburetor with Vacuum Gauge—	12. WHEELS: Includes: Remove and install wheels—Replace Grease retainers if necessary—Oil Brake linkage—Adjust wheel bearings—Tighten axle shaft or flange nuts—Tighten Wheel nuts—Report on: Loose Drums—Wheel Cylinder Bushes—Locks—Shoes—Lockage—Inflate Tires R.F. No. R.R. No. L.F. No. L.R. No. No.
13. BRAKES: Includes: Adjust—Tighten all fluid line connections—Filling of Master Cylinder if necessary—Report on: Cables—Rods—Cross Shafts—Hose—Liners—Brake Cylinders—Brake alignment—Brake operating valve—Brake Rod—Master Cylinder hose—Master Cylinder—	14. REAR AXLE: Includes: Tighten differential carrier bolts—Tighten axle bolts—Tighten U Bolts—Report on: Lubrication level—Springs arched—Flat—Spring alignment—Shackle Bolts—Auxiliary spring bolts—Gaskets—Leaks—Radiator Water—	15. FRONT AXLE: Includes: Tighten and adjust all steering connections—Align front wheels—Tighten U Bolts—Report on: Caster—Knuckle Bushings—Axle U Bolts—Springs arched—Flat—Spring alignment—Shackle Bolts—
16. CLUTCH AND TRANSMISSION: Includes: Adjust pedal—Adjust hand brake—Replace pull back spring if necessary—Report on: Lubricant level—Leaks—	17. UNIVERSAL JOINTS: Includes: Tighten Joints and Flanges—Report on: Condition—Hanger Box—	18. CHASSIS LUBRICATION: Grease chassis thoroughly—Oil or grease universal joints—Oil all brake linkage—Fill Steering housing—Hood pulley—Door Hinges—Locks—Lubricate Speedometer Cable—Oil throttle housing—Oil back end—Bushing cylinder—2 in. each end—(Cold test oil. Four test at least 90 degrees below zero.)
19. BATTERY: Includes: Test gravity 1-2-3-4—Should be 1.275. Fill with water—Clean and Tighten connections—Tighten ground strap—Tighten battery in case—Report on: Voltage—Volts. Battery Box—Cables—Connections—	20. LIGHTING SYSTEM: Report on: Horn operation—Wiring—Headlamps—Dim Lamps—Side Lamps—Dash Lamp—Tail Lamps—Stop Lamp—Spot Lamp—Lighting Switch—Clearance Lamps—Road Lamps—Stop Lamp switch—Traction Warning Plug—Receptacle—Dimmer switch—	21. FENDERS, ETC. Report on: Condition of Fenders—Running Boards—Apex—Top Covering—General condition—
22. BODY: Includes: Clean glass—Remove all grease and finger marks—Clean inside of cab with air—Report on: Glass—Door Locks—Handles—Window regulators—Wiper Blade—Hose—Gas Gauge—Air Gauge—Body pins—Springs—	23. MISCELLANEOUS: Includes: Road test vehicle Report on: Clutch, Gear, Shifts—Brakes—Operation of transmission, Horn—Quint—Gear Shift—Speedometer—Oil—Internal Noisy—Quint—Muffler—Heat Indicator—	

REMARKS:—Any items marked (O) outline in detail

For Further Remarks Use Other Side

Fig. 10—"B" Preventive Maintenance Service



Fig. 11—Failed Steering Wheel and Housing

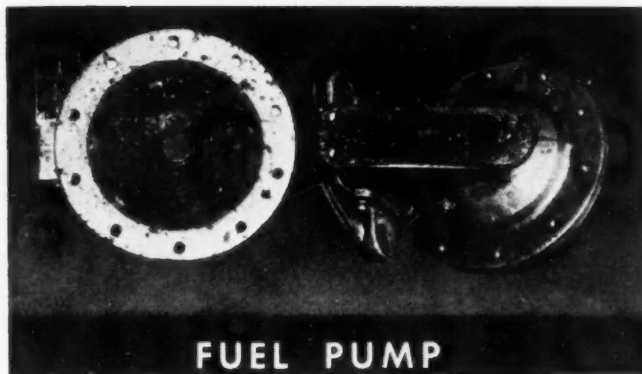


Fig. 16—Fuel Pump

ing of the engine or the exchanging of this unit for a factory re-manufactured unit will undoubtedly be necessary. This period will be established as a result of inspection and examination provided for in the preventive-maintenance service.

The second phase involves three forms: The Preventive Maintenance Service Schedule, the Fleet Maintenance Record, and the Fleet History Record. Taking them up in sequence, use of the preventive maintenance service schedule accomplishes the first principle of systematic maintenance; namely, *regularity*. Many systems fail because this first principle is treated half-heartedly. We have been called into many operations where maintenance was not effective because the availability of trucks and mechanical personnel had not been synchronized.

Let us review the preventive maintenance service schedule, shown in Fig. 22. Vehicles are grouped according to monthly mileage. The schedule is sufficiently flexible to accommodate any actual mileage grouping. The example schedules a group of vehicles operating from 1000 to 3000 miles per month. Note that mileage separates them into distinct classes and that the schedule automatically orders each truck into the shop every 1000 miles.

The schedule also provides for variations in truck-operating conditions. On high-mileage trucks where operating conditions are not severe, the maintenance superintendent may decide that, while the truck requires lubrication each 1000 miles, the complete *A* Service is only necessary each 2000 miles. A red X on the schedule signifies that lubrication only is required. Again taking a high-mileage truck, but this time assuming severe operating conditions, the use of a black X only indicates the necessity for the complete *A* Service each 1000 miles. (In Fig. 22 bold face type has been used to denote a red "X" and light face type to denote a black "X.")

The schedule is made up monthly—Sundays and holidays being blocked out—and the services are distributed through the available working days to provide a reasonably even flow of work through the shop. The "peaks and valleys" so often prevalent are leveled out, resulting in a more economical and efficient shop operation.

The second of these forms—the fleet preventive maintenance record, shown in Fig. 23—although perhaps formidable in appearance, is really very easy to maintain and to the practiced eye reveals maintenance facts instantly. It might well be termed the "heart" of the system. The pertinent items of record are: (1) Mileage; (2) date service was performed; (3) nature of the repair; and (4) mechanic performing repair.

The nature of the repair is coded in accordance with a

breakdown outlined at the foot of the form. Some operators have gone into further detail on coding, assigning the letter *A*, for instance, to all adjustments; *R* for replace; *L* for reline; etc. Therefore, 2-*A* would indicate a rear-axle adjustment, while 4-*A* would signify a brake adjustment. Others are using the form in conjunction with a flat-rate schedule, which makes it possible to assign a definite operation number to each detail of repair.

Referring directly to the three trucks whose history appears in this example, we were recently called upon to make an analysis of a fleet consisting of some 200 pieces of equipment which had been operating without any maintenance system. In a review of the completed analysis, the repair history of these three trucks was so interesting that we decided to portray it in Fig. 23.

Regular repairs are shown in black (medium-face figures are used in Fig. 23).

Repeat work is shown in green (light face figures are used in Fig. 23).

Road failures are in red (bold face figures are used in Fig. 23).

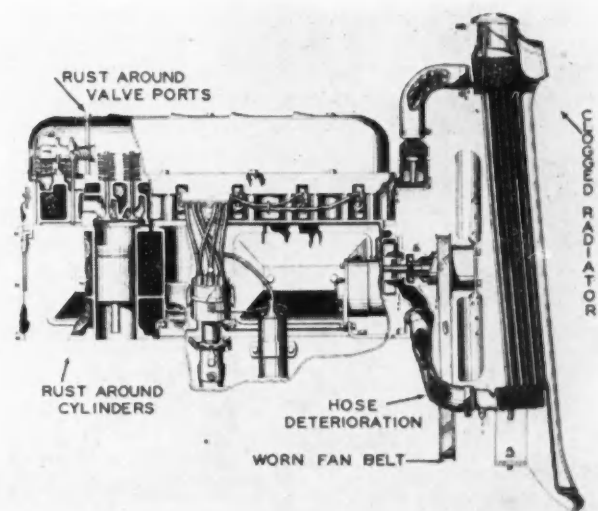


Fig. 17—Cooling System

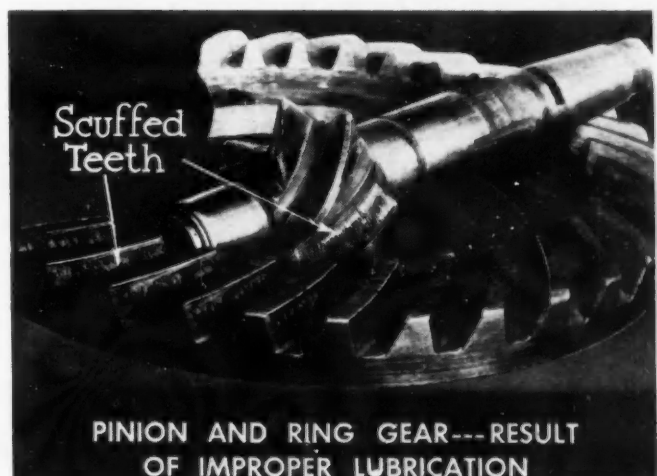


Fig. 18—Worn Ring Gear and Pinion

There were 14 repeat operations and three road failures on Truck No. 479 in four months. Discussing road failures first, we find that faulty ignition points were the cause of two of the failures and in each instance the points were replaced as evidenced by operation 7-1200. The code number (20) in each case indicates a "tow-in." The mileage accumulated between these two failures was only 3000 and, since contact-point life is considerably more than this, some other unit contributed to these failures. Very likely it was the condenser; but, in neither instance was the condenser checked or replacement made.

The third road failure occurred as a result of clogged gasoline lines (Operation 12-1230), although the vehicle was in the shop for engine tuning just a few days before (8-100).

The examples of waste which were brought to light, as the repeat work was charted, were really serious. Several units of the vehicle were involved, but the generator and water pump seemed to be the most constant offenders.

The generator first appears on Feb. 22, at which time an attempt was apparently made to adjust the charging rate (Operation 7-1820); but later it was decided to overhaul the generator, (Operation 7-2000). The repair was apparently satisfactory for approximately one month, but under dates of March 26, 27 and 28, the charging rate was adjusted and again on April 2, 4 and 5. On April 6, it was again decided to overhaul the generator (7-2000). There had been an accumulation of only 7000 miles between these two major repairs and five different mechanics had tried their hand at correcting the difficulty.

The water pump appears on March 10, (Operation 6-950); repack pump. The repacking continued at frequent intervals March 24, 27, April 10, and June 9; that is, five attempts to stop a leaking water pump in less than 90 days.

Other evidence of inefficient workmanship is found under Operation 8-100; tune engine. This was performed on March

Case 7:18-10341-MS Document 1-1 Filed 06/28/18 Page 1 of 1

General Motors Truck Company
PREVENTIVE MAINTENANCE WORK SHEET
 MODELS T-16 TO T-43 INCLUSIVE
 "D" SERVICE

[illegible]

U.S. AHS — Any items marked (C) outline in detail.

For Further Remarks Use Other Side

Fig. 19—"D" Preventive Maintenance Service

27, although the engine had received the same treatment some 200 miles previous. It was tuned again on April 19 and six days later, on April 25, the engine was again tuned, the vehicle having been run only 147 miles.

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General Motors Truck Company PREVENTIVE MAINTENANCE SERVICE SCHEDULE

MECHANICAL ☒

LUBRICATION 

MONTH October

YEAR 1935

Miles	Track No.	Route No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	Miles
1000	3																	X																
	8																		X															
	11								S						S				X			S								S				
	23																			X				X										
2000	9							U							U							U	X						U					
	14								X																			X						
	6							N		X					N							N						X		N				
	21									X		X											D						X			X		
3000	18							D							D								D						D					
	7																																	
	4		X					A					X		A							A		X						A				
	10			X										X												X								
3000	13				X																		Y				X							
	5					X																				X								
	10						X																				X							
	Total P. M. Services		/	/	/	/	/	/	/	/	/	/	2	/	/	/	2	/	/	/	/	/	/	/	/	/	2	2	/	/	/	/	/	

Fig. 22—Preventive Maintenance Service Schedule

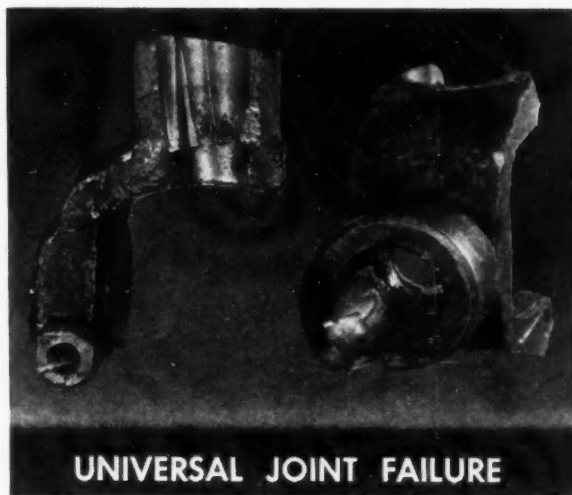


Fig. 20—A Universal Joint Failure

Let us look at vehicle No. 482. The record on this vehicle starts with a road failure on Feb. 23. The report shows that it was necessary to tune the engine and clean out the gas line (Operation 8-100 and 12-300). The following day the engine failed again, this time it being necessary to replace the contact points and condenser (7-1200 and 7-1500). This correction was apparently successful because there is no further evidence of ignition trouble. On March 10, among other operations, the battery was removed, charged and replaced (Operation 7-4150). This operation appeared again March 20 and 29. Finally, on April 6, in conjunction with a fourth recharging of the battery, the generator was overhauled (7-2000), apparently correcting the condition. The fact that the speedometer cable was out of order for a period of some two weeks is indicated by the same mileage reading for the period of April 19 to May 1. On May 6, it was necessary to adjust the clutch (Operation 5-150). This operation was performed on May 9, and again on May 12. On May 13, the clutch failed entirely, making it necessary to tow the vehicle to the garage and effect a complete repair (Operation 5-900 and 5-950).

Vehicle No. 490, a different make than No. 479, reflects a



Fig. 21—Worn Spring Shackle and Bracket

GENERAL MOTORS TRUCK CO.
FLEET PREVENTIVE MAINTENANCE SCHEDULE.

Register Make _____ Mileage _____
Type Model _____
Oil Used Daily _____
Gas Used Daily _____
Mach. Failure _____
Mechanic's No. _____

Total Mileage _____
Current Month _____
Oil in Qts. _____
Gas in Qts. _____
Miles per Q. _____
OIL _____
Miles per Q. _____
GAS _____
Miles per Q. _____

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
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1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	2	3	4	5	6	7																								

	1000	2M	3M	4M	5M	6M	7M	8M	9M	10M	11M	12M	13M	14M	15M	16M	17M	18M	19M	20M	21M	22M	23M	24M
P.M. Service "B"																								
P.M. Service "C"																								
P.M. Service "D"																								
1 Rebuild Steering Knuckles																								
1 Overhaul Front Axle																								
2 Change Differential Lubricant																								
2 Overhaul Differential																								
4 Reline Brakes																								
4 Bleed Brake System																								
4 Overhaul Master Cylinder																								
4 Overhaul Wheel Cylinder																								
5 Overhaul Clutch Assembly																								
6 Radiator and Hoses																								
6 Repack Water Pump																								
6 Overhaul Water Pump																								
6 Replace Fan Belts																								
7 Overhaul Generator																								
7 Replace Generator Brushes																								
7 Overhaul Starter																								
7 Replace Spark Plugs																								
7 Replace Points and Condenser																								
7 Replace Motor & Distr. Cap																								
8 Adjust Main & Conn. Rod Brgs.																								
8 Replace Pistons, Pins & Rings																								
8 Grind Valves																								
8 Overhaul Oil Pump																								
8 Clean Oil Pan, Oil Lines & Screens																								
8 Overhaul Engine																								
9 Replace Muffler																								
12 Overhaul Carburetor																								
12 Replace Fuel Pump																								
13 Clean, Repack & Adj. Wheel Brgs.																								
15 Adjust All Shackles																								
15 Rebuild Shackles																								
16 Overhaul Steering Gear																								
17 Change Transmision Lubricant																								
17 Overhaul Transmision																								
18 Overhaul U Joints																								
18 Overhaul Hanger or Hanger Brg.																								
19 Repl. Frt. & Rr. Wheel Grease Rot.																								

1 - FRONT AXLE, SPINDLES, ETC.
2 - REAR AXLE
3 - BODY
4 - BRAKES
5 - CLUTCH

6 - COOLING, FAN, WATER PUMP
7 - ELECTRICAL, IGNITION, LIGHTING
8 - ENGINE
9 - EXHAUST, MUFFLER
10 - FENDERS & R. BRGS.

Fig. 24—Maintenance

similar condition, the operation being fairly consistent until May 1 when the engine was tuned. The following day, May 2, the engine was again tuned. On May 3, a road failure is reflected and the carburetor was overhauled, which apparently eliminated the necessity for daily engine tuning. The brakes began to demand attention on May 22, when an adjustment was performed (Operation 4-100). Again, on May 25, the brakes were adjusted and on June 16 they were freed-up and adjusted (Operation 4-200). On June 21, it was necessary to reline the brakes and, when this was done, drum replacement was also necessary (Operation 4-400 and 4-1100).

This fleet maintenance record serves another purpose, because its study permits the gradual separation of the "Jockey" and the "Careful Driver." Vehicles requiring constant attention, traceable directly to abuse, stick out like the proverbial "sore thumb" and the necessary corrective measures can be taken, substantiated by facts. It is also possible to use this form in a slightly different manner than as shown in the foregoing. In the upper right-hand corner is a code arrangement incorporating daily gasoline and oil usage. Without sacrificing valuable information, such as we have shown in the example, fleet operators who do not maintain a separate gasoline and oil record use this code rather than the one shown in the left-hand corner of the chart, which is the one we have shown in the example.

The third and final record is the maintenance history record, shown in Fig. 24. The information recorded on this form, when used in conjunction with the previous forms, constitutes the Preventive, or "Repair Before Failure" type of main-

tenance system. Forty-three items of repair are arranged on the left-hand side of the form. They are numbered to correspond with the same code that is used throughout the system. This form is flexible, since items may be changed or added to as found necessary in a particular fleet. Opposite the repair operations are 52 heavy-columns, each representing 1000 miles of operation, thus providing a 52,000-mile history.

Previously, I associated the *B*, *C*, and *D* Preventive Maintenance Services with definite advanced mileages for purposes of illustration, but I said at that time that the items contained in these various services did not necessarily apply literally to all makes of vehicles and all types of operations. If they did, there would be no necessity for a history record such as this.

Assuming, however, that Services *A*, *B*, *C*, and *D*, are in effect at the mileage periods we previously used—namely: 1000, 5000, 15,000, and 30,000 miles—we have illustrated here how deviations from these definite mileage periods may be recorded to suit variable operating conditions. We have placed the *B*, *C* and *D* Services in their respective mileage brackets. It will be recalled that the *D*, or 30,000-mile service, included a complete brake relining; however, if because of very severe service brakes last only 15,000 miles, brake relining—front and rear—is then blocked out on this chart at 15,000-mile intervals.

In another example let us assume that new spark plugs at 10,000-mile intervals improve performance and economy. The chart is blocked out at 10,000-mile periods to care for this.

The *C* or 15,000-mile service includes valve reconditioning; but, assuming that experience has proved that 20,000 miles is

[illegible]

- | | |
|-------------------------------|-------------------|
| 11 - FRAME, BRACKETS, ETC. | 16 - STEERING |
| 12 - FUEL GAS LINES, PUMPS | 17 - TRANSMISSION |
| 13 - LUBRICATION | 18 - UNIVERSALS |
| 14 - NEW TRUCK DELIVERY | 19 - WHEELS |
| 15 - SPRINGS, SHACKLES, CLIPS | 20 - TOW-IN |

February, 1936

Fuel Consumption Problems

By Alex Taub

Development Engineer, Chevrolet Motor Co.

THE fuel consumption prevailing today is no better than it was five years ago. Higher road speeds are responsible. Cars in the hands of owners today are below potential economy between 10 and 15 per cent. Minor adjustments can correct this.

Phasing of the burn with valve and piston movement is necessary for economy. Spark-plug position relative to the whole chamber is important. Spark-plug points position inward is important. The spark-plug gap width must be worked out. Mixture "fish hooks" to determine leanest mixture that will burn without raggedness, are the yardstick.

Mixture distribution is important. The attitude of today is that a specific type of manifold does not exist. The manifold must be "tailored" to fit individual conditions. Offside mixtures out of the carburetor is real problem of distribution.

Exhaust dilution of the mixture is the handicap to clean operation with lean mixtures. Timing is the most important avenue of progress.

FUEL consumption, as such, is not often made the subject of an entire paper. We allude to it, in a hopeful sort of way; then we go on to other less elusive things. Elusive indeed, we must admit, since the fuel consumption now prevailing is no better than it was five years ago. This is so, in spite of the improvement in efficiency and in our knowledge of what is needed. This condition may rightly be blamed on smoother engines, mainly due to engine mountings, which together permit and encourage higher cruising speeds.

Net fuel consumption is controlled by part-throttle operation. A few years ago, speeds between 20 and 40 m.p.h. were those at which economy was made or lost. Today, part-throttle operation from 40 m.p.h. to 70 m.p.h. are the economy controlling speeds.

Six years ago the average miles per gallon for all cars at 70 m.p.h. was about 8 miles per gal. Cars in the lower price

range did not operate at these speeds, but averaged no better at 60 m.p.h. than the larger cars did at 70 m.p.h. As the smaller cars picked up road speed, the fuel-consumption picture changed and, in 1933, the average at 70 m.p.h. was between 9 and 10 miles per gal.

Today, with 80 m.p.h. as the average top speed, the fuel economy at 70 m.p.h. is 12 to 14 miles per gal. In fact, improvement in consumption has been made only at speeds from 40 m.p.h. up. This is a very important improvement, because, without this, present tank mileage would be lower than in the past.

The important factors in the high-speed mileage improvement are: first, that all carburetors have two operating mixture ratios, one for full throttle and a leaner one for part throttle—a difference of 1 to 2 ratios—depending on how well the carburetor man has followed through. Obviously, if at 70 m.p.h. one is using full-throttle mixture-ratios due to throttle position, good mileage is out of the question. The present day wide-range engines are operating at part-throttle mixture-ratios at speeds above 70 m.p.h.

The second important condition is heat loss. Improvements in reduction of heat rejection invariably aid high-speed economy a great deal more than the lower speeds. Just exactly why, I do not know.

The third favorable condition for improvement is the fact that the wider range of our engines permits operation at these high speeds with high manifold vacuum and a throttle-plate position that, contrary to opinion, is an aid to distribution in present-day engines. Add to this the lean part-throttle mixture-ratio—and we have an ideal.

Specific economy is always greatly improved at high speed by thermal improvement, since the increase is in percentage of work done. The nearer we approach full-throttle operation for level-road operation, with part-throttle manifold-vacuum and distribution, the more ideal this should be unless offset by over-rich mixture, usually provided in fixed-jet carburetors for wide-open work. Part-throttle mixture-ratios permit satisfactory constant-speed operation, at high speeds.

Normal Economy Methods.—It is safe to say that any job on the street today is performing from 10 to 15 per cent below its intended mileage in fuel. Set the tappets to their recommended clearance, the spark-plug gaps to the proper distance, and set the spark advance up to at least borderline detonation, and you can pick up 10 to 15 per cent in fuel—provided of course that the job is reasonably loose and the valves are seating.

This is not very scientific and therefore is uninteresting to

[This paper was presented at the Annual Meeting of the Society, Detroit, Jan. 16, 1936.]

research engineers; yet the fundamentals of good road economy lie in these adjustments. In fact, unless these adjustments are thoroughly determined, there is little or no chance for good economy.

High Compression.—High compression is a common approach to economy. The limiting factor is roughness. We have special fuels that will wash out detonation, but nothing to offset roughness. Therefore, in selecting cylinder-head material, consideration should be given to such materials as will give the maximum economy for the minimum compression ratio. This takes into consideration materials that absorb heat and dissipate power and economy, making necessary further increase in compression ratio to offset the loss by absorption.

If L-head engines must have high heat conductivity for suppression of detonation, why not have this arranged for locally and have normal heat transfer at all other sections of the chamber?

The area of the initial burn, or the first one-third of the volume to burn, is the area most sensitive to heat loss. It is exposed to high temperature for the longest time, and, as we have been told, this is the area of maximum temperature. For these reasons the area should be protected from heat loss if we are interested in economy.

The improved expansion ratio resulting from higher compression is unquestionably important in improving economy, but the lowering of the exhaust dilution may be more important.

Revolutions Per Mile.—Lowered revolutions per mile is another normal avenue of attack. Overdrives have made their appearance and, for the speed ranges at which they operate, the road economy is improved. Larger engines that permit lower revolutions per mile throughout the range appear to be a more satisfactory solution. The cycle for this development has started, and in the interest of smooth operation, durability, and economy, we hope the movement will grow.

Volumetric.—Improvement in volumetric, usually considered a means of wasting fuel, is an excellent aid to fuel economy, particularly at the speeds above 50 m.p.h., or near full-throttle operation. Heat loss improvement is partially responsible, and lowered pumping losses are likewise to be credited.

Friction.—Lowered friction is a known avenue of approach to improved economy. However, it is seldom traveled. New cars are still delivered into the hands of the public too tight for economic use, and are usually out of adjustment when broken in. This means that, with some cars, the full potential in economy is never reached by the owners. Engines are built tight when early wear is expected, and must be offset.

Tin plating of iron pistons introduced a new durability factor; by reducing bore and piston wear, at a ratio of 3 to 1.

Permanence of piston fits, with a piston that does not collapse under load, permits reasonable original fits. If you do not have to provide against excessive looseness between piston and bores, close original fitting pistons are unnecessary. Between an original fit of 0.0015 and 0.002 in. we have the difference of 2 miles per gal. at 20 and 30 m.p.h., and a mile or so at 40 m.p.h. and up.

Spark Setting.—This is a most important factor in the matter of economy, and is different for full throttle and part throttle. Since tank mileage depends on part-throttle operation, it is necessary that the spark be adjusted to suit the load—or work—done. Fortunately the manifold vacuum follows

the work done reasonably well and gives us a medium to interpose between the full-throttle spark-advance and that necessary for part-throttle operation.

Part-throttle operation may require from 5-deg. to 20-deg. additional advance to obtain the maximum economy. Quite often it is difficult to obtain the range required since, when sufficient advance is obtained, to give good results up to 60 m.p.h., sufficient over-advance is left to make the engine rough at higher speeds. A mechanical cut-off for the vacuum is the best answer. Ports in the throttle shaft can be arranged to cut vacuum in and out where desired.

Thus so far we have discussed the normal attack on the problem of fuel consumption. We all know that we may have all of the foregoing as good as possible, and still have an unsatisfactory result from an economy standpoint. I am of the opinion that we have been looking over the head of our problem.

Lean Mixtures

Fundamentally, we are interested in burning lean mixtures without the usual raggedness that limits lean operation. In 1934 fuel experts advised us that our operating mixtures were too lean; they were fearful that we would get into trouble with such leanness.

I am avoiding evaluating the mixture ratios here because I would have to give such values in three different ways; on the flow bench, on the dynamometer through air-consumption calculations, or by exhaust-gas analysis. There is a 2 to 3-ratio discrepancy between the three methods.

Getting back to lean mixtures, we experienced no trouble in 1934 and have since further leaned out our mixture almost a whole ratio, improving our operating economy accordingly.

The point I wish to make is that no one can sit on the outside looking in and know whether a mixture ratio is too lean or not, for a given engine. Obviously, the leaner a mixture can be and give complete combustion, the better the operating economy is. The leanest mixture to burn is the ideal to shoot at, and whatever that ratio is will give satisfaction.

We have not the slightest idea as to what mixture ratio represents the leanest that will burn, because we have not stopped long enough to find out.

The limiting factor is not the mixture ratio that leaves the carburetor on the way to the cylinders, but the mixture within the cylinder. The exhaust gas present in the combustion chamber is the real limiting factor for road economy. The percentage of non-burnable gas in the cylinder under part-throttle condition is so high, as compared to full throttle, that it is a wonder that it burns at all.

The evidence pointing to this limiting condition is clean cut. First, there is the condition where the engine will not respond to treatment and is ragged with a relatively rich mixture, until the spark plug is moved to a more favorable position. This condition does not prevail because the small amount of live fuel is chasing around like a comet within the chamber, successfully missing the plug points. It does not burn because the local area at the points is overloaded with a non-burning mixture, or too much exhaust gas which has not been swept away from the points; but you move the plug to a new position and the job may begin to react to treatment.

Pushing the plug around to find a burnable mixture is not a recommended practice, since this leads to a compromise with engine roughness which is a practice of the devil. The chamber shape at the plug should be designed to eliminate exhaust traps.

But, once having established a location, mixture "fish hooks" should be made at road loads and speeds for various positions of the points inward. Too far inward means hot plug-points; too far outward always gives a poor result in the leanest mixture that will burn. Move the plug inward by counterboring $1/32$ -in. deeper for each step. The gain will flatten off and a point is selected for the minimum position inward for best economy.

The mixture "fish hook" that best serves this purpose is a constant speed and load, varying the throttle opening when necessary and using a predetermined constant spark advance for each speed. The miles per gallon over the manifold vacuum will give the picture.

Moving the plug inward places the plug through the layer of exhaust gas that clings to the walls and in the plug pocket. Experience teaches that the plug points should be $3/32$ -in. below the wall intersection, and never in a pocket.

Spark-Plug Gaps

With the plug position determined, it is now necessary to determine spark-plug gaps. New mixture "fish hooks" should be made, increasing the gap 0.005 in. at a time from 0.020, until a gap is reached at which maximum economy prevails. If this should be more than 0.045 in., one is in trouble; since, to date, more gap than this may lead to bad cold-starting, or trouble with the plugs at high duty. The fact that excessive plug gap is required is mute evidence that the engine is a bit sick and must be given other treatment which we will discuss later.

Plugs are graded according to their operating temperature. It is a pity the grade numbers are not standardized. Experience indicates that the hotter the plug is, the leaner the minimum mixture may be. Perhaps the indications may point to a grade of plug that would not give the necessary full-throttle durability. It would be well worth the effort to change the water circulation around the plugs to permit the use of a so-called "hot plug". There is 1 mile to $1\frac{1}{2}$ mile per gal. difference up to 30 m.p.h., between a No. 7 and a No. 10 plug.

There is a reason for this fussing with plugs. It is an important reason. The efficiency or effectiveness of the burning charge is a matter of proper phasing with the piston movement and valve opening.

We do not question for a moment the fact that retardation of the spark will mean a loss in economy. Obviously, this is because we are burning out of phase with the moving parts.

Manipulation of the source of ignition supplements the proper spark timing by eliminating ignition lag, which may easily represent, in time lost, many degrees in spark advance. The overall time of burning is so short that any lag becomes sizable in degrees of crank travel, and lag may occur due to the low burnability of the mixture. Locating the plug without pockets helps in that it does not make a bad situation worse. Proper gaps give a longer or more virulent start. The hotter plug may aid by preparing the local mixture for ignition. They all work toward hurrying inflammability to insure that the maximum amount of heat shall be converted.

Super-advancing the spark position for part throttle does not hurry the burn, but it does ignite early enough to provide time for complete and effective inflammation.

Mixture Distribution.—Mixture distribution has always been relied upon to obtain good fuel consumption. However, the industry's fuel consumption is not particularly good. Does that mean that the mixture distribution of the industry's en-

gines is not good, or does it mean that it has very little to do with good economy? The answer is, of course, that, important as is even division of the fuel, it is not the neck of the bottle.

Getting back to mixture distribution, a few years ago it was the custom of engineers to "pass the buck" of mixture distribution, along with all problems of carburetion, to the carburetor man. He was an outsider and could not talk back; but he did do a lot of complaining when he could not be heard. He was expected to fix a bad manifold and poor timing, and rotten ignition with the carburetor. Today, years later, the carburetor man is not complaining. The design engineer does not have to tell him that the carburetor is to blame for most of the bad distribution. The carburetor man knows this only too well.

Today we realize that the principal division of the mixture occurs at the tee of the manifold. The distribution is controlled by how the fuel comes out of the carburetor. The manifold must be tailored to fit the carburetor, provided it is within the realm of possibility.

The position of the nozzle wherefrom the fuel first enters the carburetor is important. Air holes in primary venturis must be placed to push the fuel to one side or the other, depending on how the mixture behaves in the tee.

Manifolds

There are many more detailed carburetor tricks that must be considered. Corrections are made in the manifold to suit the prevailing conditions. There is no specific design of manifold that may be considered better than others.

Manifolds are the result of offsetting existing errors that start in the carburetor. However, it is better to deal with a definite error than to have to deal with a partial error that may be one thing at one speed and something else at another speed.

The manifold should be designed for equal air flow to all cylinders with a minimum overall restriction. The rest must be a matter of tailoring. Square turns on the outside, that at one time were considered necessary, we know today are useless because of bad flow-lines. Today we provide flow lines through our manifolds for wide-range operation and mechanically correct for wet distribution. In fact, since wet fuel on the end turns behaves in a definite manner at all speeds, it is possible to make a definite correction that holds equally true for all speeds.

We accumulate the wet fuel and guide it where we want to in the quantity indicated. Types of manifolds that will bounce fuel into the air stream will be right for some speeds and not for others, and they do not lend themselves to tailoring; consequently they are hard to fit to carburetors with off-side mixtures.

A great deal more work has been going on in controlling the condition of the mixture homogeneity as it enters the manifold than in devising manifolds. The result has been progress.

Yes, the carburetor man is carrying on today, doing it with the carburetor, although he tailors the manifold as he goes along.

Exhaust Dilution.—Exhaust dilution is the obvious obstacle to inflammability, and represents the proper avenue of attack. We must reduce the exhaust gas present in the cylinders at the time the exhaust valve closes and the intake valve opens. For years, the industry has known that the overlap between intake opening and exhaust closing was responsible for bad

idling. We were concerned when the actual overlap was about 25 deg. Today, many engines operate with an intake-exhaust overlap of from 60 to 80 deg.

Do the engines idle? No, they certainly do not. The necessity for tappet adjustment as an aid to economy is obvious, because close tappet clearance with present-day long ramps gives almost impossible conditions of valve timing. We tell ourselves that overlap is necessary for high-speed performance; and who drives below 10 m.p.h. anyhow?

Well, overlap may help high-speed performance, but it is by no means the only type of timing that will do the trick. This overlap actually promotes a high accumulation of exhaust gas, allowing exhaust to flow back due to differential pressure of exhaust and intake. When exhaust valves are too small—and this is general—exhaust gas under pressure is actually trapped in the cylinder, which not only lowers the overall volumetric efficiency but adds to the exhaust dilution.

When the valve timing is such as to permit a good idle, we can be sure that the leanest mixture that will burn has been improved. Lowering the exhaust content will make excessive spark gap unnecessary, and the super advance for part throttle will be relatively low. This makes for smoothness and stability, and certainly for spark-plug durability.

Looking at the fuel-consumption problem as the business of being able to burn lean mixtures, we see many opportunities for work. Surely, ways and means of diminishing exhaust dilution promise much, with good economy and a decent idle as the reward.

Spark gap and its effect on economy has always seemed intriguing. The fact that the wide gap does so well with a bad mixture has always seemed a promise of better things.

Some years ago, during an investigation where it was necessary to set the plug gaps at 0.045 in. to make the engine hit at light loads, we decided to check the result by reducing the gap and doubling the voltage. We went from 6 to 12 volts, and the raggedness disappeared with the plug gaps set at less than 0.020 in. In other words, we need a more contagious ignition; whether we get it by increasing the exposure such as by long gaps, or by using more virile germs such as by increasing the voltage.

Some day we will have a controlled variation in voltage interconnected with a vacuum. We may then have just the potential required at the plug points needed for ignition under all conditions of load. This is not an invitation to ignore exhaust dilution, but is offered as a threat as to what might happen if we do not do something about exhaust dilution.

Thus we see there are "no rabbits in the hat," and therefore none can be pulled out where fuel economy is concerned. We must go to work and along prosaic lines, but to work we must go, and I am sure the results will be gratifying.

I have offered no charts and no quantitative data. Such information might becloud this issue, as each engine will require different treatment and the data will be different. I do, however, hope at some later date to have the opportunity of comparing notes with you, as fellow workers, on fuel consumption.

Airplane Design Needs for Unskilled Pilots

THE motion of an airplane in flight is a combination of the pilot's efforts and the natural uncontrolled airplane motion. We have a small amount of information regarding the motion of an airplane while flying with free controls but in spite of 20 years of experience in training pilots we have no definite information as to how the novice pilot reacts to the normal situations encountered in flight. Except for marginal cases, we have no data regarding the control time lag to expect from experienced or inexperienced pilots; we have no criteria of what the normal allowable misuse of the controls should be; and consequently we have no concise relations connecting the pilot and his airplane.

Outside of producing a pleasing shape, very little is done about the form of the airplane where it affects flying characteristics. As a vehicle, the airplane is unfortunate in that its form to a large degree dictates its motion, and in some cases, the aerodynamic requirements for a desirable motion may not be compatible with the current idea of what looks good. Moreover, since there is no stable basis of good appearance, any such indefinite criterion delays progress. More rapid development would result if we could set up numerical yardsticks that we thought desirable and build the airplane to meet those conditions regardless of the resulting appearance. In addition, there is no question but that in the long run the machine having the best characteristics would be considered the best looking.

Much remains to be done in building up a system of numerical criteria of design as affecting flying characteristics. The design conditions now used are minimum requirements

and correspond to the minimum requirements of the Department of Commerce. The following are the design requirements about which there is general agreement:

1. The variation of rolling, yawing, and pitching moments with angle of yaw and pitch should be in the direction to produce a restoring moment.
2. There should be no reversal of control motion under any condition.
3. There should be a definite increase of control force with control position.
4. The controls should be properly coordinated.

The writer suggests that in addition to the usual minimum design requirements, attention be paid to the following characteristics when an airplane is designed particularly for amateur pilots:

1. The natural period of any oscillation should not be less than 15 sec.
2. Any oscillation should damp to less than one-half amplitude in less than one period.
3. For the landing condition, the added parasite drag to steepen the flight path should be variable with control position.
4. The rate of change of stick force with stick movement should be increased beyond the value normally obtained in the basic design.

Excerpt from the paper "Smart Airplanes for Dumb Pilots" presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1936, by Otto Koppen, Massachusetts Institute of Technology.

The Measurement of Engine Knock by Electro-Acoustic Instruments

By Neil MacCoull

The Texas Co.

and

G. T. Stanton

Electrical Research Products, Inc.

THIS is another chapter in the development of a quantitative knock-measuring apparatus. The first part of the paper describes tests in which the knock sound was picked up by a microphone located a few inches from the cylinder head of a test engine.

Oscillograms were taken of the knock disturbance, and measurements were made of the energy in various frequency bands. The latter study included the effects of L-head and overhead-valve design, as well as some work with cylinder heads of iron, aluminum and bronze.

It was found that the C.F.R. standard engine develops a pronounced peak in knock energy at frequencies between about 6000 and 7000 cycles per sec., while the L-head design of this same engine had an additional peak between 3000 and 4000 cycles per sec.

It was found also that a pronounced background of engine noise develops at frequencies below about 2500 cycles per sec.

A brief account is given of the application of these principles to the measurement of knock in cars on road tests.

The second part of the paper deals with a special microphone installed so as to be in direct contact with the gases in the combustion chamber which are at the seat of the detonation disturbance.

It has been found that the use of filters which cut off low frequencies to varying degrees has a pronounced influence on knock rating, and may be of considerable importance in the development of a routine knock-measuring procedure which will give better correlation than the method now in use of individual fuels with values obtained in cars on the road.

THIS paper describes the results of measurements of engine knock conducted with electro-acoustic apparatus.

The work was undertaken for the purpose of improving the correlation of fuel-knock ratings obtained in the laboratory with those obtained in motor cars under road conditions. In 1932, when these experiments began, there existed a lack of agreement between road and laboratory methods which, in the minds of some, was serious. Why did they not agree? The answer to this question obviously required a consideration of those physical factors which cause knock, and

an investigation of the differences in behavior between car and laboratory engines.

The usual simple explanation of the cause of knock in cars is a sudden pressure rise at the end of the combustion cycle caused by the explosion or spontaneous ignition of the last portion of fuel to burn. The energy developed by this sudden pressure change is transmitted through the combustion-chamber walls, and to some extent, other parts of the engine, causing an audible "ping" or other sound. The reduction of this sound, which is objectionable to motorists, has been, and still is, the main purpose for all antiknock research. It appears obvious from an acoustical consideration of this phenomenon,

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 18, 1935.]

that the audible external sound will be modified by the nature of the engine construction. Each individual design, and the same design under different operating conditions, will have a variability in sound transmission and damping factors for different frequencies of sound. Thus, certain types of engine design produce sharp high-frequency or "tinkling" sounds, others a well-defined "ping" and others range to a muffled "thumping" sound which is difficult to differentiate from ordinary engine noises. It is, of course, obvious that, for a given fuel, the engine design also alters the intensity and nature of explosion and the resultant frequency chain. Also, there are indications that fuels of different type may, under identical engine conditions, have different knock characteristics, although their quantitative measurement by the tentative standard laboratory tests, indicates equal octane ratings.

Previous Development

It is rather interesting to trace briefly the history of knocking work, for this paper is simply another chapter in this development, and by no means the final chapter. The earliest record we have found in literature to the existence of knock as a fuel characteristic was in a patent issued in 1915. It was through the research in this field that the modern high compression has become possible. Since the loudness of the knocking sound at the driver's ear limited the compression or type of fuel, the obvious original method of testing fuels was to gage their relative loudness. The obvious necessity for an instrumental method¹ resulted in the development of the familiar "bouncing pin" in 1921. Experience with this device showed the necessity for standardizing the engine in which such measurements were made because of the changes in the apparent knock value of fuels when compared in different engines.

In 1928, the Cooperative Fuels Research Committee agreed

¹ See S.A.E. TRANSACTIONS, vol. 17, part 1, p. 126, 1922; "Methods of Measuring Detonation in Engines," by Thomas Midgely, Jr. and T. A. Boyd.

² See S.A.E. TRANSACTIONS, vols. 25 and 26, p. 464, 1930-1931; "Effect of Sound Intensity on Knock Ratings," by Harry F. Huf, J. R. Sabina and J. Bennett Hill.

³ See *Journal of the Institution of Petroleum Technologists*, p. 513, 1932; "The Strobe Phonometer," by Carpenter and Stansfield.

⁴ See *Electrical Engineering*, May 1931, p. 342; "A New Meter for Noise Analysis," being an indicating meter for the measurement and analysis of noise, by T. G. Castner, E. Dietze, G. T. Stanton, and R. S. Tucker.

⁵ See *Bell Laboratories Record*, May, 1935, p. 267; "A General Purpose Frequency Analyzer," by T. G. Castner.

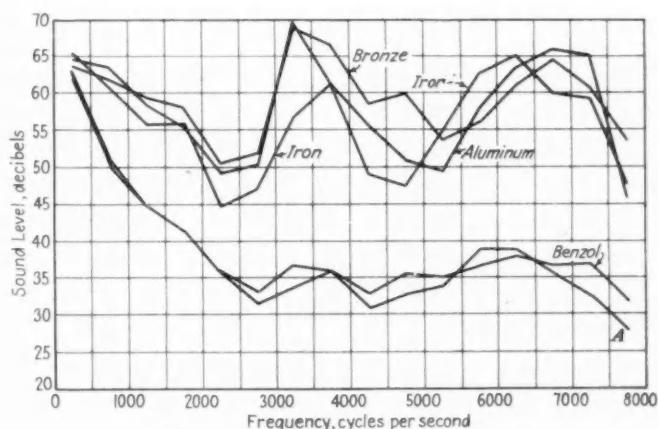


Fig. 1—Sound Spectra from an L-Head Engine for Various Head Materials

The curves for benzol and fuel A, were for knock-free operation, and indicate the level of mechanical engine noises.

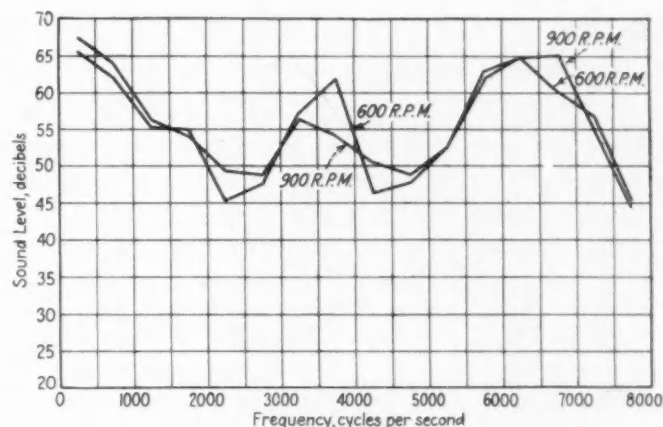


Fig. 2—Sound Spectra from an L-Head Engine Indicating that Engine Speed Has Very Little Effect

on the elements of a single standard engine, which is now in almost universal use for this purpose. Unfortunately, however, this still did not prove a complete answer to the problem of correlation with road performance. Considerable experimentation has been and still is being done in an attempt to develop a set of test conditions and procedure which will provide adequate correlation between laboratory knock ratings and relative knock loudness ratings of fuels in modern cars.

Prior to the development of the present tentative A.S.T.M. method, using the bouncing pin, several laboratories had experimented with audibility methods, feeling that this was fundamentally the correct way to obtain correlation. More recently Huf, Sabina and Hill² improved the audibility method by substituting sound-measuring apparatus for the ear of the observer. This method was further elaborated by Carpenter and Stansfield³ in England. In April, 1932, a series of road tests using a conventional form of sound meter was made in which the increase in total engine noise with detonation was measured. While this method was interesting in that it substituted quantitative values, it is obvious that the sensitivity in this method is very low since knock exists for a very short period of time and, therefore, the increase in total noise is quite small even for very loud knock.

External Measurements

Definite work on the tests herein reported was commenced in 1932, using a sound-level meter⁴ and a frequency analyzer⁵ in an attempt to determine some of the physical characteristics of audible knock. The sound meter measures the total sound-level at the microphone. When used in conjunction with the frequency analyzer, the intensity in any desired band of frequencies may be independently measured; thus, determining the "frequency spectrum" of the noise. With the ability to select certain frequencies at will, it was believed that the sensitivity of these measurements of knock could be materially increased.

The first data were taken on the early L-head type of C.F.R. engine using four fuels: one straight-run and one highly cracked, each of 75 octane number; two blends of straight-run having 90 octane rating by the addition of lead and benzol, respectively. As no material difference was observed in the characteristics of the straight-run and highly cracked fuels, their results are averaged in the data to be shown.

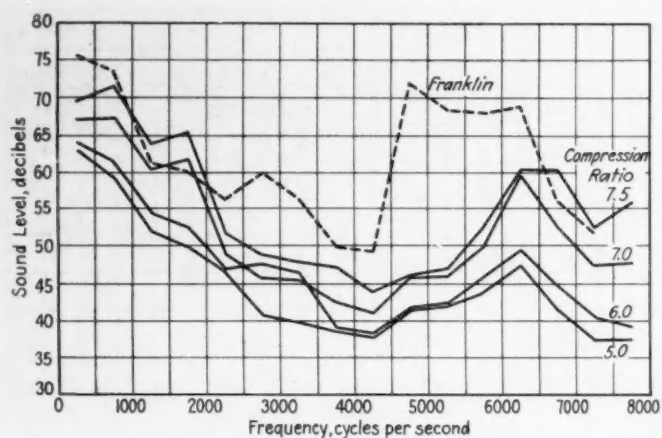


Fig. 3—Sound Spectra from an Overhead Valve, Variable-Compression Engine, With Various Knock Intensities Obtained by Raising the Compression Ratio

Fig. 1 shows three sound-spectra which are typical of these tests, and which were obtained with cylinder heads of iron, aluminum and bronze, respectively. The spread between any of the upper curves and the two lower curves represents the increase in sound intensity due to detonation for the frequency specified, since the lower curves represent the 90 octane fuels, which were non-knocking. Examination of these curves indicates the following conclusions:

- (1) Knock causes an increase in sound energy at all frequencies measured.
- (2) For this particular engine, the increased noise due to knock is especially noticeable in the frequency band between 3000 and 4000 cycles and in the band between 6000 and 7000 cycles per sec.
- (3) Cylinder-head material, all other factors being equal, has only a minor effect on the spectrum.
- (4) The increase in sound energy at low frequencies due to detonation is small, while for this range the general background of engine and other noises becomes increasingly prominent. It was therefore concluded that, for this particular engine, the sensitivity of the apparatus would be increased by a filter eliminating frequencies below about 2500 cycles per sec.

The effect of engine speed on spectrum characteristics was investigated, using the same engine and fuels, the result being shown on Fig. 2. The changes due to the different engine-speeds appeared to be negligible.

As an indication of the change in characteristic due to engine design, runs were made with the same fuels on the conventional variable-compression-type A.S.T.M.-C.F.R. engine. This engine, of course, has a compact overhead-valve combustion-space as distinguished from the elongated space in the L-head type previously tested.

The solid lines of Fig. 3 indicate the sound spectra for knocks of various intensities as controlled by the compression ratio. The general similarity of frequency distribution for rather wide changes in knock intensity is apparent. By comparison with Fig. 1 of the L-head-type engine, a radical change is found in that the prominent 3000 to 4000-cycle peak previously noted, is absent but the 6000 to 7000-cycle peak re-

mains prominent. For additional comparison, the dotted line shows the spectrum of knock from a Franklin air-cooled cylinder fitted to a C.F.R. crankcase. This engine produced a very sharp, clear "ping" which is manifested by a broad peak between 4500 and 6000 cycles.

To determine the effect of changes in operating conditions, runs were made on the A.S.T.M. variable-compression engine with the same fuels under conditions similar to those of Fig. 3, except that the jacket temperature was 375 instead of 212 deg. fahr. These curves, shown in Fig. 4, are distinguished by a sharp trough in the vicinity of 4000 cycles.

Additional measurements were made of the external sound-wave by the use of a rapid-record oscillograph⁶. This device, together with its associated amplifiers, has, in the range from 30 to 10,000 cycles, a substantially flat response. With it, the actual amplitude of the pressure wave is photographed on moving sensitized paper, permitting determination of the "time lag", amplitude and duration of the knock sound and, if desired, the frequency of the amplitude change.

In the oscillograms presented in Figs. 5 and 6, the light vertical lines represent time spacings of 0.001 sec. Fig. 5 shows a continuous record of two successive detonations which, because of its length, is here reproduced in sections. This is a record from the L-head engine operated at 600 r.p.m. The lower trace indicates by a sudden jog, the time of the spark, which may also be seen in the upper curve. The spark advance was 21.5 deg., or about 0.006 sec. before dead-center. Each vertical line represents 3.6 deg. of crankshaft rotation. It will be seen that detonation began 0.0075 sec. after the spark, or 1.5 deg. after dead-center. The amplitude of the detonation disturbance is fairly rapidly damped, the final stopping point being masked in general noise. It is, however, substantially negligible after 0.008 sec., or 29 deg. of crankshaft rotation.

In Fig. 6, the top curves show the important part of two similar oscillograms considerably amplified. In this record the background of engine noise is more prominent. The bottom curve is a record from the air-cooled cylinder previously mentioned, and indicates clearly a material change in characteristic sound from that of the water-jacketed cylinders.

The result of the development to this point was to substantiate in general the originally accepted belief that the engine design exerted a strong influence on the nature and characteristic of the audible-knock sound. It also indicated

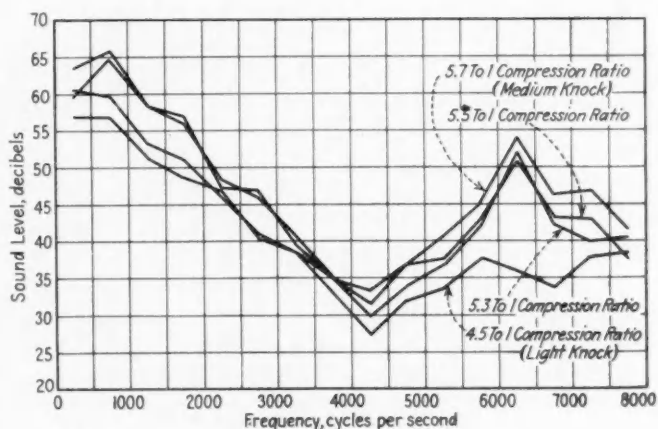


Fig. 4—Effect of Changes of Knock-Intensity at High Jacket Temperature

⁶ See *Electronics*, August, 1931, pp. 70-71; "A Rapid-Record Oscillograph," by A. M. Curtis and I. E. Cole.

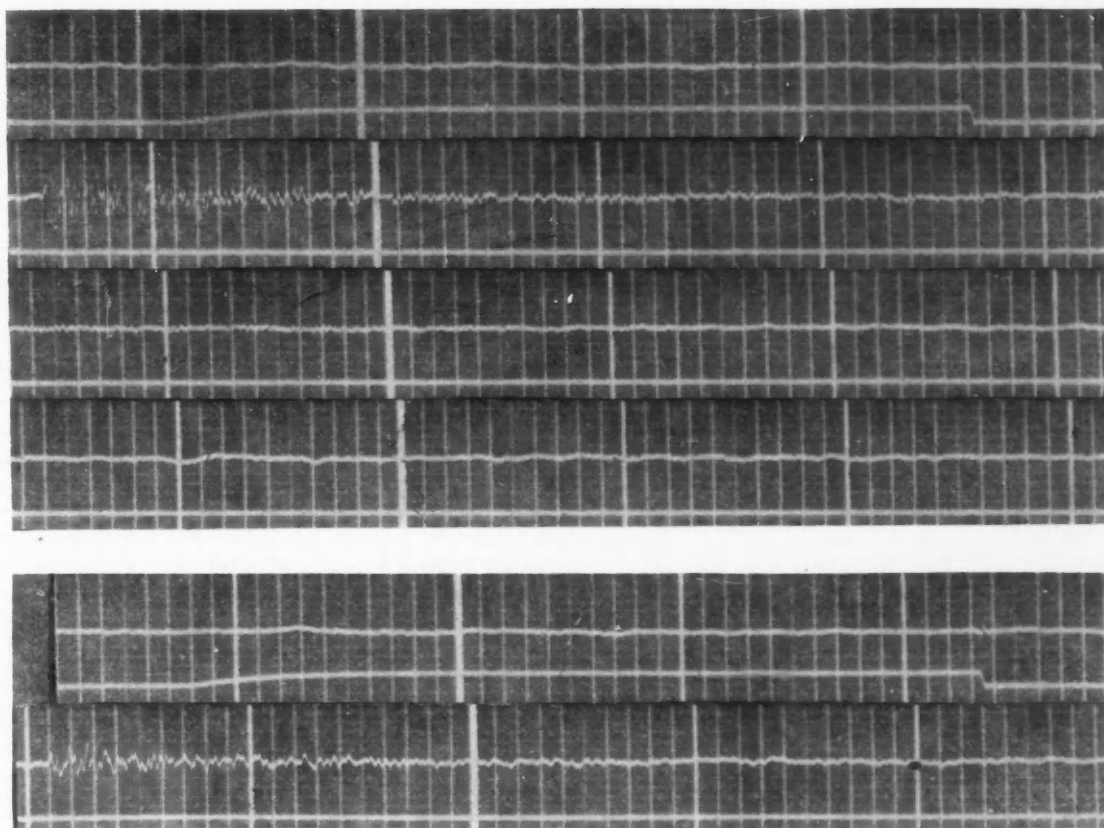


Fig. 5—Oscillogram Showing a Continuous Record of Two Successive Detonations

the degree of influence exerted by certain changes in operating conditions and in the material employed in the engine head for the specific engines tested. It is especially significant that the location in the spectrum, of peak energy, by which an observer defines knock, varies widely from engine to engine.

An interesting summary of the results obtained is shown in Fig. 7. The curves in Group *A* of Fig. 7 show the effect of increasing knock intensity by spark advance, when measured with the sound meter alone, the effect on amplitude as determined by the oscillograph, and the effect on time lag between time of spark and origin of detonation. In Group *B* is shown the correlation between the intensity recorded by

the sound meter, and amplitude recorded by the oscillograph. The group here shown indicates reasonably consistent and similar results by both methods. The curves in Group *C* show the effect of compression ratio on knock intensity as indicated by overall measurements with the sound meter. The uniform relationship between sound-meter reading and compression ratio is clearly brought out, as well as the material change in slope of this relationship by change in the jacket temperature.

Road Meter.—It is, of course, necessary to obtain as accurate knock readings in cars on the road as it is in the laboratory in order to assure correlation between laboratory engines and car tests. Attempts were made to measure knock values in cars by methods very similar to those described in this paper, but experience showed the same limitations. In addition, these methods were believed far too laborious for practical road testing. For this reason a different form of device was developed which conveyed the sound of knock from a microphone located near the engine to the observer's ears by means of an amplifier and head phones, all apparatus being selected to reproduce faithfully the knock sound. The degree of attenuation required to bring the knock to the "threshold" of audibility, provided comparative measurements of the intensity of knock under different conditions. This device was demonstrated at the last C.F.R. Uniontown-Hill test and seemed to function satisfactorily in the hands of those who used it sufficiently to become familiar with it. A more detailed description may be found in Appendix 1.

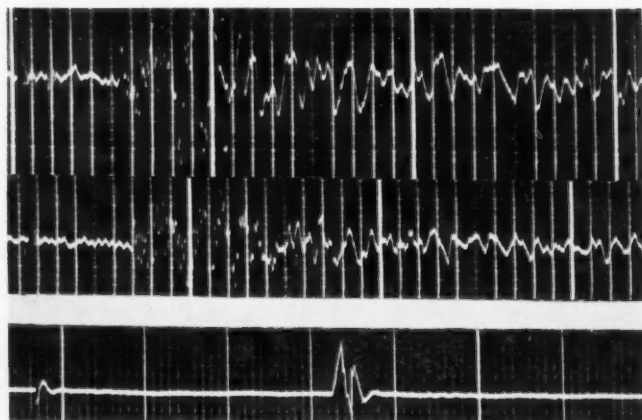


Fig. 6—Oscillograph Records

The top records show the important part of two oscillograms similar to Fig. 5, amplified. The record below these is from an air-cooled cylinder.

Internal Measurements

After reviewing all of the preceding work, it became apparent that if measurement could be made inside the cylinder

instead of outside, more information could be secured as to the original source of the knock sound. Among the advantages offered by such a design are:

(1) The elimination of interference from engine or other noises, complications introduced by room acoustics, variability in microphone position, and the like

(2) Elimination of the alteration in the sound caused by passing through cylinder walls

(3) Direct measurement in reproducible units of the factors associated with the individual fuel behavior.

In the design of this equipment, consideration was given to three general types of microphones; namely, Piezo-crystal, Electro-dynamic and Electro-static. While each of these possesses certain advantages and disadvantages, the electro-static or condenser-type was ultimately selected, primarily because of the simple and rugged construction obtainable and the fact that reproducible units could be produced easily. A full discussion of the design problem and the equipment produced may be found in Appendix 2.

The apparatus with which the majority of the following data was taken consisted of a small condenser-microphone with a diaphragm 0.045 in. thick, mounted flush with the in-

ner surface of the cylinder head. The voltage generated by movement of this diaphragm was amplified, rectified and measured in arbitrarily designated sound units on a highly damped ballistic meter. From the preceding experiments, it appeared obvious that the lower frequencies should be suppressed to eliminate certain engine noises and thus increase the sensitivity. In an attempt to determine what portion of the spectrum might be suppressed without losing any of the essential characteristic of the knock, an extensive series of high and low-pass filters was inserted in the circuit, permitting the selection of a large number of combinations of frequency bands up to 3000 cycles per sec.

Oscillograms.—Before describing the quantitative measurements made with this apparatus, a picture of the physical nature of the knock disturbance will be given by means of oscillograms made by a cathode-ray oscillograph associated with the engine microphone and its amplifiers. This will be an aid in interpreting the data which follow.

Fig. 8 shows a series of oscillograms for knockfree combustion, with various "high-pass" filters; that is, the low frequencies were cut off to progressively higher limits. It will be seen that the knockfree combustion disturbance, while quite pronounced, is much reduced by the filters, and is practically eliminated by filters cutting off all frequencies below about 750 cycles per sec., or higher. This figure should be compared with Fig. 9, which is similar except that the latter oscillograms were taken with a fuel which detonated severely under the same engine conditions. It will be seen that a decided disturbance remains with the "high-pass" filters, which cut off all frequencies below 750, 1500 and 3000 cycles per sec. respectively. Since we know from Fig. 8 that practically no non-knocking disturbance passes filters cutting off at 750 cycles or higher, oscillograms F, G, and H of Fig. 9 must be records of the knock itself.

In analyzing the oscillograms of Figs. 8 and 9, it should be borne in mind that the amplifiers were designed for use with an indicating instrument, and no particular effort was made to eliminate the phase distortion which is introduced by audio-transformers. For this reason an analysis of the oscillograms, especially those with no filter, is very difficult. For instance, Fig. 10 shows an oscillogram taken under the same conditions as for A in Fig. 8, except that an amplifier with resistance coupling was used instead of one with transformer coupling, and the amplification on the vertical scale was greater. It will be seen that the part of the curve following the first swing below the line is radically different. If a study of knock and combustion phenomena are to be made from oscillograms, particular attention must be made in the design of all electrical circuits, to avoid phase distortion which fortunately has practically no effect on indicating instruments, like the knock indicator supplied with this apparatus and used to collect the subsequent data presented in this paper.

Knock Measurements.—The first readings taken with the knock-measuring equipment, using various filters, made it apparent immediately that the comparative ratings of some fuels were altered by changing the portion of the spectrum which was measured. One result of using high-pass filters in the measuring circuit is to suppress the effect of lower-frequency pressure-changes, which are associated with the power cycle, thus giving proportionately greater weight to the rapid changes initially caused by detonation. This method then appeared to offer a means of controlling the amount of

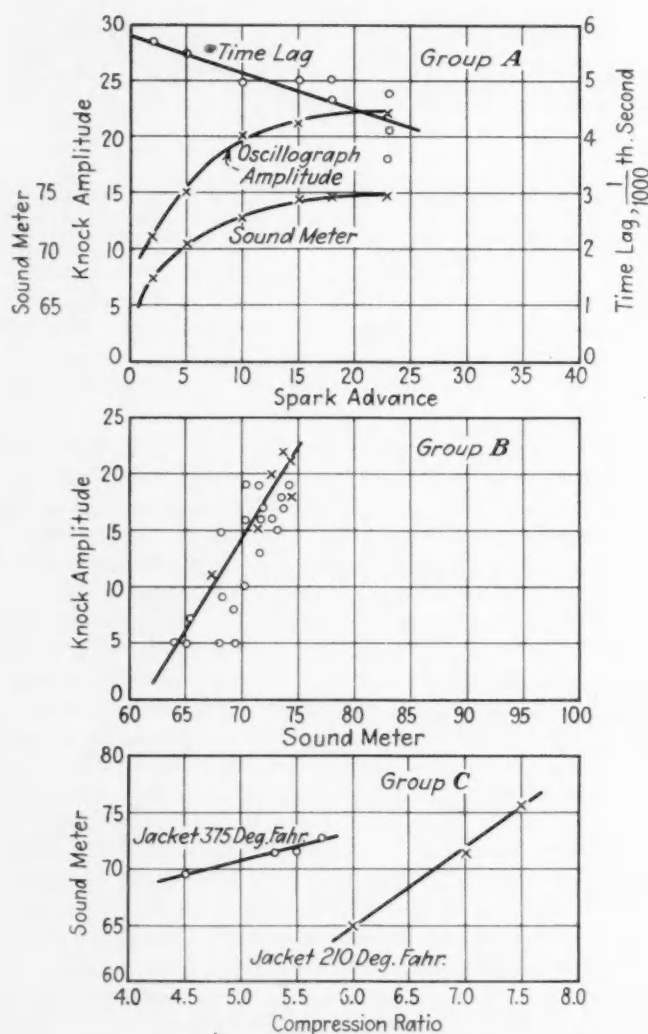


Fig. 7—Summary of the Results Obtained from Outside Knock Measurements

weighting given in the results between the pressure changes associated with normal burning and with detonation.

Since the bouncing-pin is believed to be affected primarily by the maximum velocity initially attained by the diaphragm, and the microphone equipment is affected by the maximum pressure amplitude, correlation of these should be obtained using a filter which permits such partial evaluation of the normal power-cycle as does the bouncing-pin, in addition to the effect of detonation. Since the ears of an observer have their maximum sensitivity at approximately 2000 cycles per sec., better correlation with apparent loudness should be obtained using filters having a higher-frequency cut-off.

The extent to which these assumptions have been realized is partially summarized in Fig. 11. These curves show a pronounced change in knock ratings for some fuels by control of the frequencies used, the lower knock ratings agreeing well with bouncing-pin values, while higher readings taken with a higher-frequency cut-off agree better with average ratings in the cars which we have tested.

Considering this figure in detail, it shows antiknock values of several gasolines when measured with various high-pass filters in the measuring circuit. Except for the knock-measuring apparatus, all details of the A.S.T.M. method were closely adhered to. The gasolines coded from 1 to 7 are commercial motor gasolines picked up at service pumps. Fuel *AA* is a highly cracked vapor-phase product of 70 octane number, and Fuel *BB* is a benzol blend of the same nominal octane value. The results with an iso-octane-*n*-heptane blend, and natural gasoline are designated by name.

(1) On each curve a circle indicates the filter cut-off that gives correlation with the antiknock value determined by bouncing-pin under the A.S.T.M. method. It is obvious that,

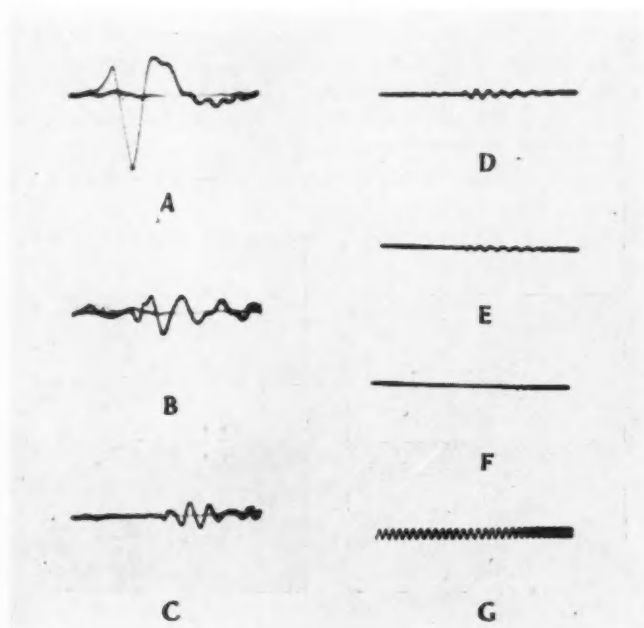


Fig. 8—Oscillograms of Knock-Free Combustion with various "High-Pass" Filters

A: No filter B: 125 C: 250 D: 375
E: 500 F: 750 cycles per sec.
G: is a timing wave of 1000 cycles per sec.

The 750-cycle high-pass filter practically eliminates the combustion disturbance, and slightly higher frequency cut-offs eliminate it completely.

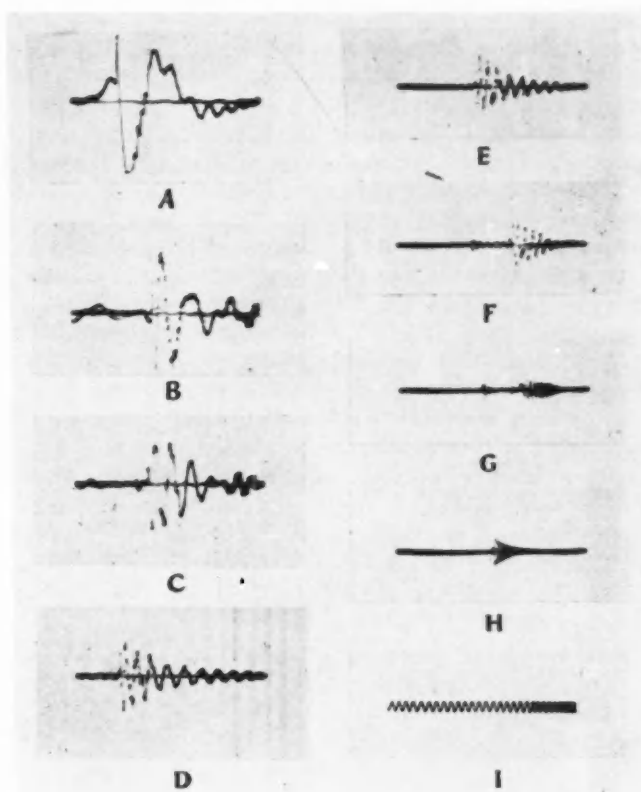


Fig. 9—Oscillograms of Knocking Combustion under Similar Conditions to Those of Fig. 8

A: No filter B: 120 C: 250 D: 375
E: 500 F: 750 G: 1500 H: 3000 cycles per sec.
I is a timing wave of 1000 cycles per sec. The knock wave persists in F, G, and H where Fig. 8 indicates the combustion disturbance to be filtered out.

The disturbances just to the left of the knock wave in F and G indicate the spark timing.

with all frequencies suppressed below about 375 cycles, a fair correlation with the present method is obtained.

(2) Following the curves of the commercial fuels, as the low frequencies above this value are progressively eliminated, it is found that in every case the antiknock value increases above that normally obtained by the A.S.T.M. method. This is particularly true of Fuel *AA*, the vapor-phase product.

(3) Conversely, the octane-heptane blend and natural gasoline fall rapidly in rating. The same is true, to a lesser extent, of the benzol blend.

(4) While time has not permitted the complete road testing of all fuels here shown, extensive tests on Fuels *AA* and

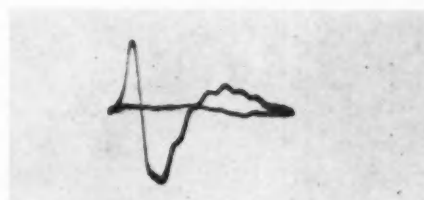


Fig. 10—Oscillogram of Knock-Free Combustion, Similar to Fig. 8-A, except for Reduction of Phase-Distortion Due to Amplifier with Resistance Coupling Instead of Transformer Coupling

No. 5 have been made on several 1935 cars under test procedure similar to that used at Uniontown. The average results of these tests are shown by crosses. An improved correlation with laboratory results, using 1500-cycle high-pass filters, is indicated.

The significance of attempting to correlate laboratory ratings with car ratings, through control of the frequencies measured, may be associated with two factors: First, the design of the engine, which affects the relative amount of high and low frequencies produced, thus indicating the selection of a particular portion of the frequency spectrum to simulate the effect in a particular car. Second, a difference in rate of pressure rise for various fuels, affecting the relative amounts of high frequencies produced by a given fuel in any engine.

The ratings given in Fig. 11 are, of course, comparative, and the first step in analyzing such data is to learn how the actual knock energy of a fuel varies with the different filters. This is shown in Fig. 12 for three basically different fuels, blended to have the same antiknock value (70) by the A.S.T.M. method; the cracked and benzol blends, Fuels *AA* and *BB* used previously, and a straight-run blend of reference Fuels *A-3* and *C-8*. This figure includes the effect of knock intensity also, as it was varied by the compression ratio from 4:1, which was too low for an audible knock, to 6:1, where the knock was so severe as to be near preignition. The following points will be seen to be of interest:

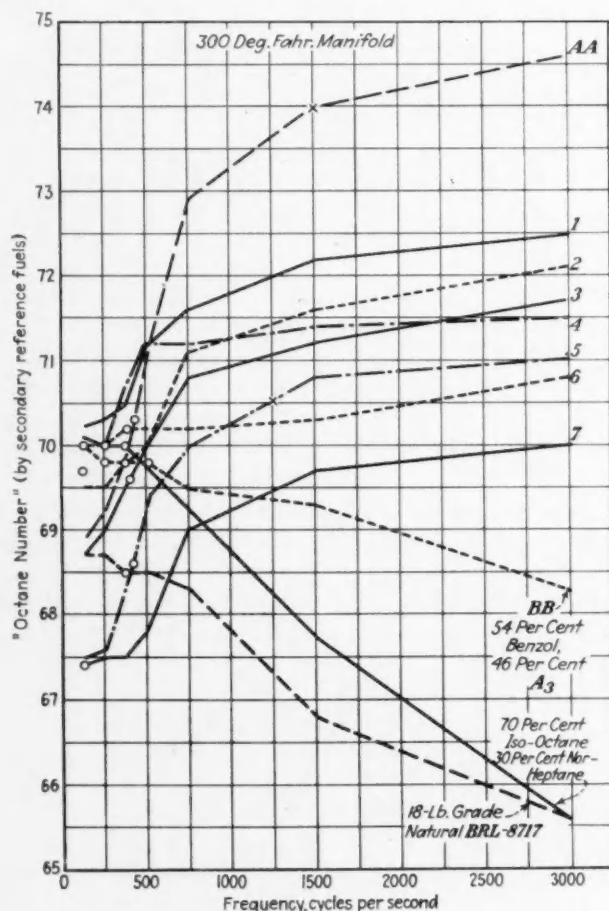


Fig. 11—Changes of Knock Ratings with Various High-Pass Filters

Correlations with bouncing pin—A.S.T.M. data indicated by O—with road data by X

(1) There is rapid falling off of total sound energy as the low frequencies are cut off.

(2) The left half of each family of curves shows no noticeable effect in form as the knock intensity is changed by means of increasing the compression ratio. The only effect of compression ratio on this portion of the curves is to displace them upward, which is probably due to the increased power being developed.

(3) All three fuels give substantially the same meter reading at any compression ratio, until frequencies up to about 500 cycles per sec. have been eliminated. As the frequency cut-off is raised still further, a wide spread begins to develop between Fuel *AA* and the other two, the spread increasing with compression ratio.

From similar curves, obtained with two blends of the reference fuels *A-3* and *C-8*, having antiknock values of 64 and 70 octane units, Fig. 13 was prepared to show the effect of frequency discrimination on "sensitivity". The sensitivity of the apparatus is defined in our work as the number of scale units per octane difference in fuels. It will be seen that the sensitivity increased steadily as the cut-off point was raised in frequency up to about 1500 cycles. By raising the cut-off still further, little improvement would appear to be gained.

The study of the effect of compression ratio and its resulting change in knock intensity was extended to include data such as shown in Fig. 14, which includes families of fuels made up with Fuel *A-3* as a base, and given increased antiknock values by additions of Ethyl Fluid, benzol, or Fuel *C-8*. The antiknock values by the A.S.T.M. method are indicated on the respective curves. The large circle on the 70 octane fuel at 5.3 compression ratio represents the knock intensity standardized by the A.S.T.M. for ratings with the bouncing-pin. The reason for the strange hump in the curves at compression ratios below 4.2 is uncertain, and may not be found on other engine designs. It was assumed to be associated in some way with the variable-compression mechanism, and the geometry of the valve-gear design which is intended to maintain constant valve clearance regardless of the compression ratio.

The blends with Ethyl Fluid and with Fuel *C-8*, having the same A.S.T.M. knock value, gave practically identical curves, and are therefore not shown separately; but the higher-percentage benzol-blends fell below the others, as shown by the dotted curves. In other words, the cylinder microphone appears to rate benzol blends higher than does the bouncing-pin. This is interesting because one of the conclusions of the C.F.R. road tests at Uniontown was that benzol blends were rated higher in cars than by the A.S.T.M. method.

Curves similar to Fig. 14 were made with other filters, and it appears that the principal effect of higher high-pass filters is to straighten the curves.

Fuel *AA* was found to have a smaller slope with all frequencies than the *A* and *C*, or the *A-3* and benzol blends of similar A.S.T.M. antiknock values. This means that if Fuel *AA* were matched with a straight-run *A-C* blend at a light knock-intensity, it would knock less than that blend at a high knock-intensity.

Mixture Ratio

Another startling result from the use of filters was their effect on the mixture ratio required for maximum knockmeter

reading. This was first revealed by the sound of the knock with the benzol blend. When making evaluations with the bouncing-pin, or with the cylinder microphone and all low frequencies, a benzol blend is generally characterized by a dull thud. But when the mixture ratio was adjusted to give maximum reading with the cylinder microphone and the high-pass filters, it took on the characteristic sharp high-pitched sound associated with other fuels. This subject was further investigated by taking meter readings as the mixture ratio was varied over a considerable range, as shown in Fig. 15, for Fuels *AA* and *BB*. Raising the cut-off frequency required richer carburetor settings for maximum scale readings, especially in the case of the benzol blend *BB*. Arrows are shown on this chart which indicate the carburetor settings for maximum bouncing-pin readings (the usual procedure for the A.S.T.M. method); for maximum power-output by the engine; and for maximum temperature on the "thermal plug" used by the U. S. Army in measuring antiknock values of aviation gasolines.

Mixture ratio is thus an important factor in rating the knock value of a fuel. For instance: Using the 750 high-pass filter and a carburetor setting of 47, in Fig. 15, both fuels match, as they do by the bouncing-pin. But either at richer settings, or at those settings giving maximum meter readings, the benzol Fuel *BB* produces considerably greater knock. This is not new knowledge⁷, but is brought up again because use of various high-pass filters with the cylinder microphone forces this question into the open again.

What mixture ratio should be used? When the A.S.T.M. established the use of that mixture ratio giving maximum knock, it was not known that instrumentation might change this ratio. Since the prime object of laboratory knock ratings of gasolines is to determine their relative antiknock performance in cars on the road, one should look to the mixture ratios of cars on the road for the answer. Our experience is that practically no car operates with mixtures on the lean side of maximum power, and most of them operate well on the rich side; at least 10 per cent rich when a car knocks the hardest. Thus, the ratios established by the bouncing-pin may be too lean to represent car practice, while the ratios determined by the thermal plug would seem to be about right. Also, it seems distinctly questionable to reset the test-engine carburetor for every fuel tested, since cars now in production are generally provided with fixed jets and all fuels ordinarily used are metered by the same jet. It would seem reasonable that laboratory routine testing should resort to a similar procedure. For example: A carburetor could be set for a flow rate which will give a mixture 10 or 20 per cent richer than that giving maximum power with a 70 octane blend of standard reference fuels, and fuels could then be rated at the resulting mixture ratio.

A modification somewhat of this type was made to runs such as shown in Fig. 11 (in which the carburetor was set for maximum knock on each fuel with the 250-cycle high-pass filter), and readings were then taken with other filters but without changing the carburetor setting. As a result, the response of Fuel *AA* was not greatly different from that of Fig. 11, but the benzol blend *BB*, instead of falling 1.7 octane units, rose an almost equal amount; while the 3-octane-unit

drop of the natural gasoline was reduced to a drop of about 2 units.

Conclusions

(1) Successful antiknock evaluations require two simultaneous investigations; ratings in cars, and ratings in the laboratory. Both should be measurements of the same physical phenomena, preferably by similar methods or a controllable weighting of the variables between the methods, should be provided. Correlation between these two lines of investigation is of value only if similar accuracy has been secured in each:

(A) Equipment and a method for measuring antiknock values of fuels in cars by the electro-acoustic method have been mentioned.

(B) Considerable detail has been given to tests made with a cylinder microphone for making knock ratings of fuels in a laboratory test engine. It has been shown that by the use of filters various degrees of correlation with either road or standard laboratory data can be obtained. For instance:

(a) Cutting off frequencies up to at least 3000 cycles per sec. raises the antiknock ratings for most fuels, the rating for one fuel (*AA*) being raised five and one-half octane units, yet a blend of iso-octane and n-heptane dropped four and one-half octane units (rated by secondary fuels).

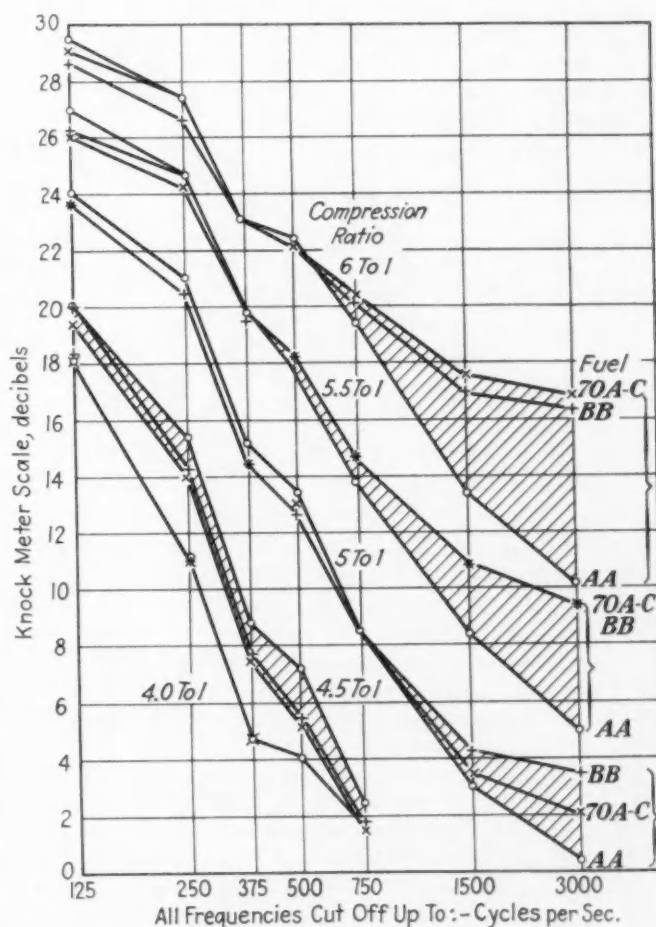


Fig. 12—Changes of Knock Intensity with Various High-Pass Filters

No appreciable difference is shown between fuels until frequencies below about 500 have been eliminated.

⁷ See S.A.E. TRANSACTIONS, vols. 25 and 26, p. 459, 1930-1931; "Influence of Carburetor Setting and Spark Timing on Knock Ratings," by John M. Campbell, Wheeler G. Lovell, and T. A. Boyd. See also S.A.E. JOURNAL, June, 1935, p. 222; "How Engine Conditions Vary Knock Tendency," by J. M. Campbell.

(b) Cutting off low frequencies increases the sensitivity of the apparatus. It also results in enriching the fuel-air mixture at which maximum scale reading is obtained, a practice more nearly in line with road procedure.

(c) Cutting off frequencies below about 375 cycles per sec. gives antiknock values in fair agreement with those obtained by the A.S.T.M. bouncing-pin method.

(d) Cutting off frequencies below 750 cycles eliminates all effects of the normal gas-pressure changes during the power cycle which are independent of detonation.

(e) Cutting off frequencies below about 1500 cycles gives fuels antiknock values which are in improved agreement with average evaluations of those fuels in cars.

(f) It seems possible that accurate correlation between cars and laboratory test engines may require different frequency cut-offs to be used for various classes of cars, dependent on their engine design.

(2) Use of an oscillograph in conjunction with the cylinder microphone gives a picture of the gas-pressure changes within a cylinder. The effects of normal combustion can be separated from detonation effects because they are characterized by much lower frequency. Examination of the characteristics of normal knock-free combustion gives an excellent means of studying "combustion roughness."

(3) The data presented in this paper have been given as a progress report and indicate trends in this investigation. The authors feel that the progress achieved so far gives good promise of being an important step in reaching the desired objective of all antiknock measurement; an accurate correlation between antiknock values obtained in the laboratory with those experienced in cars on the road.

Appendix 1

Road-Meter Design.—In planning the design of an instrument for determining quantitatively knock ratings of fuels in cars on the road, consideration was first given to the factors which define knock. These are, in general, the intensity, the frequency composition, the duration and the rate of occurrence of the knocking sound.

While a direct measuring device is desirable, it was obvious that since the duration of knock sound is very short and the intensity not greatly above that of the general background of noise from the engine and other sources, the sensitivity of such a method, using usual root-mean-square instruments with some time-integrating power, would be

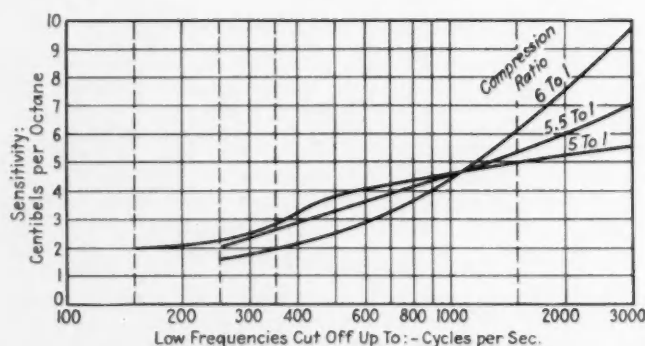


Fig. 13—Effect of Frequency Discrimination on "Sensitivity"

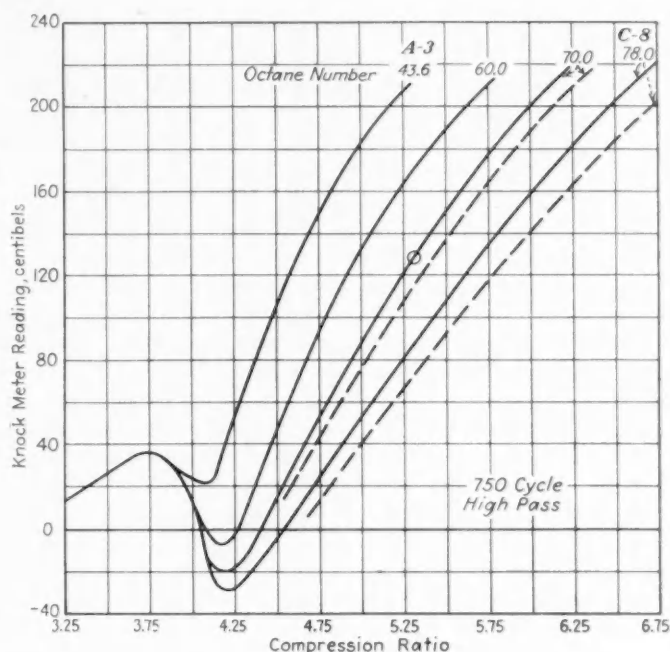


Fig. 14—Effect of Compression Ratio on Scale Reading for Fuels of Various Antiknock Values

Solid lines are for blends of secondary reference fuels A-3 and C-8—dash lines for blends of A-3 and benzol.

relatively low. Considerable experimentation was done with stationary engines and with cars operated on the chassis dynamometer utilizing an instantaneous peak recording device. Since the peak factors of the detonation sound are quite high as compared with the other forms of engine noise, some improvement in sensitivity was obtained. However, this device was not considered entirely satisfactory for actual road use without considerable additional development.

Another method of improving the sensitivity was the selection of the particular peak frequencies developed by detonation through some form of analyzer device. In addition to being an extremely laborious method suitable only for research purposes, it was obvious that the frequency spectrum would be different for every engine and, to some extent, differ with different fuels. This principle was discarded as offering only slight improvement and being relatively impractical for road work.

A trained observer is capable of concentrating his attention upon a particular sound and recognizing it in the presence of a relatively high background of dissimilar noises. Physiologically, the hearing mechanism performs with a discriminatory sense based upon the intensity, frequency composition, duration and rate of occurrence. To this extent, we need go no further in search of a device for the selection of the knock sound. The single limitation of this method is the inability to rate the intensity, or loudness, in quantitative fashion or to retain for any extensive period, an impression of relative loudness. Individuals, however, have a sharply defined limit of audibility for any given sound. This factor has long been used in scientific investigation where it is desired to give quantitative value to sound intensity or loudness.

Accordingly, a simple device was constructed to take advantage of the observer's ability to select and identify sound of

knock and to provide an instrumental means of quantitatively rating the intensity. For this purpose a microphone is located under the hood of the car at a convenient point. Associated with the microphone, with suitably shielded leads, is a calibrated attenuator, amplifier and a set of headphones. For successful use the apparatus must have a very uniform response throughout a wide frequency range, particularly in the upper portion of the sound spectrum so that the observer hears the knock sound free from distortion. The ratings may be made under any of the usual road procedures with the modification that the observer hears and identifies the knock through the headphones.

After identification, the attenuator is adjusted to the point at which the knock is barely audible. The degree of attenuation required to reduce the loudness to threshold is a direct measure of the relative knock intensity. Upon substitution of a comparison fuel, the procedure is repeated. From the experience of the authors and some limited experience at the last C.F.R. Uniontown-Hill test, it appeared that the accuracy of a single reading in this fashion was favorable with averages of a large number of observers made under similar conditions. In addition, it does not appear necessary to establish such close bracketing with reference fuels for the reason that, with quantitative indications of the intensity for two bracketing fuels and the test fuel, interpolation may be safely made over a wider bracket.

One limitation of this device exists in the case of attempts to rate fuels which are knocking with extreme intensity. In this case the observer may be confused in locating the threshold point by leakage of sound around the receiver caps. This limitation may be partially overcome by the use of sound-insulating caps on the ear pieces. This limitation does not appear serious in any event, since interest is more generally centered upon knocking conditions far less severe than any which might cause interference in the foregoing manner.

For the convenience of the operator, the apparatus may be arranged to contain the amplifier and its associated batteries in a small case, which may be placed upon the floor of the car. One model of the device contained in the amplifier case three selective filters which suppressed differing amounts of low frequencies. Some operators found the use of these filters advantageous in certain cars in partially suppressing the engine noise without changing the quality of the knocking sound. The attenuator may be mounted in a small box on the data board customarily used, thus placing the intensity control at a convenient position.

Appendix 2

Engine-Microphone Design.—In designing a microphone for measuring the knock disturbances within an engine cylinder, three general types were considered:

- (1) Piezo-crystal
- (2) Electro-dynamic; (a), moving coil⁸, and (b), moving armature
- (3) Electro-static

A consideration of the properties of these three different

types and some amount of experimentation with each led to the early exclusion of the Piezo-crystal type because of the relatively complicated and massive moving parts which are required to protect the crystals from temperature effects, and which limit the high frequency response. Of the two remaining types of microphone, there are advantages and disadvantages peculiar to each. The electro-dynamic type gave a relatively high output for a given pressure change, and at the same time, due to its low impedance, had considerable freedom from inductive pickup from electrical circuits about the engine. It did, however, have several defects which it was believed more than offset the advantages. Among these were the susceptibility of the magnetic circuit to variable flux

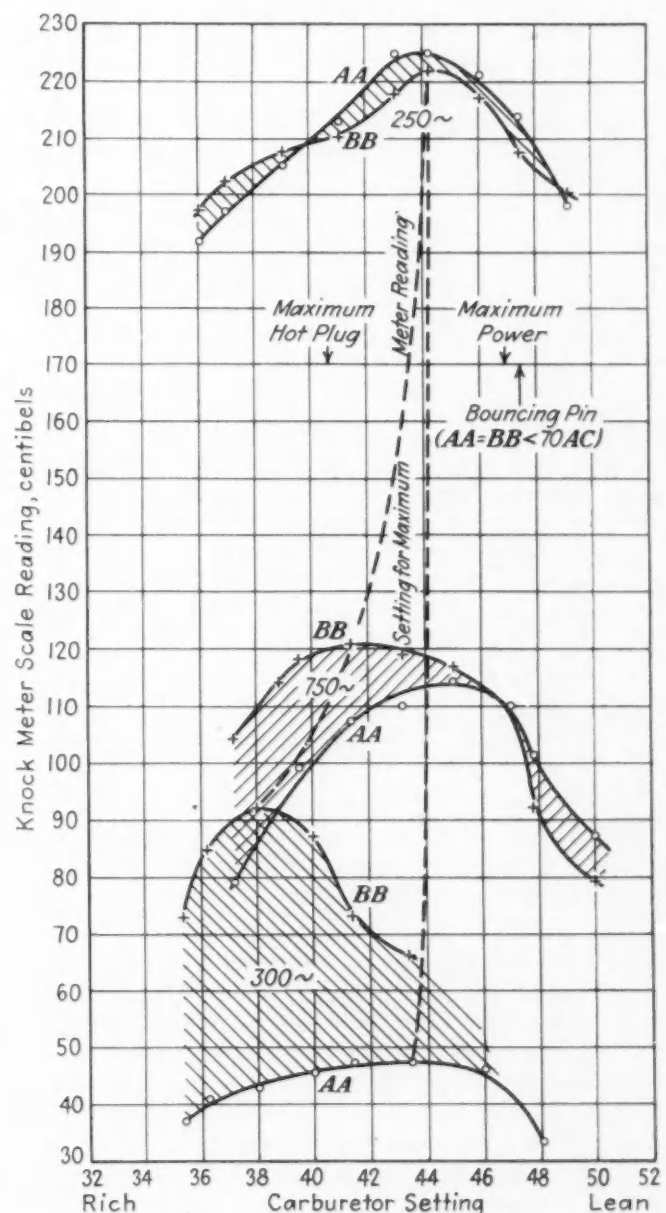


Fig. 15—Effect of Carburetor Setting on Knock Intensity, with Setting Indicated for Maximum Engine Power. Hot Plug Temperature, and the Normal Setting obtained from Maximum Knock Meter Reading on the A.S.T.M. Bouncing Pin

⁸ See S.A.E. TRANSACTIONS, 1922, part 1, p. 194; "Photographic Recording of Engine Data," by Augustus Trowbridge.

density due to changes in the relative spacing of the pole pieces from temperature variations and extraneous vibration effects, except for conditions of saturation. In addition, a relatively complicated mechanical structure was required, and the probability of some means of cooling was indicated to protect the insulation of the coil. Furthermore, reproducibility of results among microphones of this type would require extremely high precision in manufacture and assembly.

It was believed in the case of the condenser microphone that the advantages of this type over its disadvantages, outweighed the desirable factors of the electro-dynamic type and it was consequently selected for use. The decision was based in general upon the following factors:

Advantages.—(1) Readings directly proportional to pressure amplitude within the range of knock frequencies encountered. By means of adjustable filters, the amplitude of any desired frequency or band of frequencies may be measured.

(2) Since this microphone does not require special cooling of its parts, it introduces no unknown thermal conditions in the engine cylinder.

(3) Since the design of the microphone permits interchangeability of the integrally machined tips containing the diaphragm, the sensitivity of the microphone can be readily changed to meet special conditions by reducing either the thickness of the diaphragm or the air gap.

(4) Simple, rugged mechanical structure. The electrostatic or condenser microphone inherently functions with no moving parts. Its operation depends on the flexure of a simple diaphragm. This permits an electro-mechanical design yielding a high fidelity of response through a wide range of frequencies. It also permits extremely rigid construction avoiding the effect of extraneous vibration. By proper control of expansion coefficients, it is relatively simple to maintain calibration through large temperature ranges. With only two dimensions requiring close tolerances, a high standard of reproducibility may be maintained.

Disadvantages.—(1) High sensitivity to inductive pick-up. This requires good shielding of the ignition system. In actual practice, a commercial shield for spark plug and lead wire has been found adequate.

(2) Relatively low output and susceptibility to change of capacity in leads. This requires the association of a preliminary amplifier immediately adjacent to the microphone.

The microphone actually developed for these tests consists of a stem or shank of heavy seamless steel tubing. One electrode of the microphone is a flat metal plate mounted upon a rod of vitreous material selected for its dielectric properties and expansion coefficients. The conductor to this electrode passes through the center of the dielectric rod. This rod is firmly held in the exterior shank of the microphone under extremely high radial pressure. The tip of the microphone is machined from a solid block of heat and corrosion-resistive steel. This is externally threaded to fit the aperture of the test engine with the diaphragm flush with the interior surface of the cylinder head. The tip is screwed and pinned to the shank of the microphone, insuring accurate and permanent alignment and spacing of the two electrodes. The preliminary amplifier is contained in a small case attached directly to the end of the

microphone shank. The single vacuum tube and associated parts are arranged on a unique form of shock-proof mounting which eliminates from the readings any possible microphonics from this source. Conventional shielded wires are used to associate this amplifier with the final measuring or indicating devices.

Initial measurements with this microphone were made with a device for measuring the instantaneous peak amplitude of the pressure. With the particular apparatus employed for this purpose, it was not convenient to average the peak amplitudes of successive detonations. Due to the fact that there was considerable deviation between the peak amplitudes attained in successive cycles, this method was discontinued, pending development of a more suitable form. A more convenient indication was used for the data given herein, namely a ballistic type of meter which integrated the total energy over an extensive period of time. This had the advantage of giving a steady single reading, but reduced the absolute sensitivity of the device inasmuch as the duration of the energy contributed by detonation is very small compared with the total time integrated.

The design of the amplifier required to raise the energy level from the microphone and the rectifier needed for measuring purposes, required unique consideration. The wave shape of the electrical facsimile of the pressures in the cylinder is extremely peaked; that is, the ratio of the instantaneous peak energy to the root-mean-square value of the wave is great. Since all usual types of indicating devices function upon root-mean-square value and the rating of an amplifier's power handling capacity is also based on these values, special precautions were of necessity taken to insure undistorted transmission of the peak value. In the final design the undistorted power capacity is approximately fifty times the root-mean-square value required to give full-scale deflection of the indicating meter. In this particular device, the frequency response of the amplifier and associated parts is essentially linear between 50 and 10,000 cycles per sec. and is free from resonant circuits, the time constants of which might cause erroneous readings when subjected to high peak factor inputs. The amplifier is designed to operate entirely from 110-volt 60-cycle current, and also supplies power required for the microphone and its associated amplifier.

When the microphone is used in association with an oscillograph or other device to study the variations in pressure caused by normal combustion, the frequency response may be made such as to yield diagrams of the rate of primary pressure changes. This arrangement was not made for the accompanying oscillograms.

A means of relative day-to-day calibration was devised by adaptation of an air whistle in which the diaphragm of the microphone formed the closure of the whistle tube. By careful control of the air pressure a quite uniform pressure amplitude was developed at the base of the microphone. Readings conducted in this fashion from day to day over an extended period of time during which the equipment was in continuous daily operation, showed very small fluctuations. This indicates the suitability of this form of device in determining knock value of a fuel not only in relationship to another fuel but in terms of absolute magnitude under standard conditions.